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ANNUAL MEETING PAPERS

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XII January, 1923 No. 1

Chronicle and Comment

Reduced Railroad Fares

A LARGE percentage of the members of the Society are entitled to purchase transportation to the Annual and Chicago Meetings at a material reduction in cost. The saving in many cases will be sufficient to pay a member's dues for 1 year. Be sure to read the announcement on p. 131.

Gear Noise

TWO of the papers read at the Production Meeting contributed the experience of production engineers who have made careful investigations into the causes of gear noise. The paper by the Packard engineers, H. J. Crain and J. Brodie, advanced the theory that the anti-friction bearings supporting the gear shafts were fitted without sufficient clearance for an oil film, and that an increase of the clearance resulted in an appreciable reduction in the noise produced by the gears when in motion. In the discussion printed on p. 23, another authority corroborates the Packard experience with increased bearing-play, but advances a new explanation for the noise reduction that appears to be a logical one. Other valuable opinions are presented in the discussion of K. L. Herrmann's paper on p. 26.

Transportation

WHERE $1 is spent to haul a ton of freight by road wagon from New York City to Newark, N. J., a distance of 9 miles, that dollar will haul the same freight as far as Plainfield, N. J., or 24 miles over good roads, by motor truck. It will haul the same freight as far as Baltimore by steam railroad. It will haul this freight from New York City to the City of Washington, and back through canals and inland waterways. It will haul it across the Atlantic Ocean by ship.

Every time a producer in the West ships 2 carloads of grain to Liverpool more than 1 carload is absorbed in transportation charges.

Copper and zinc are now being shipped from Montana 800 miles by rail to the Pacific, thence through the Panama Canal and over 7000 miles by sea to Perth Amboy, N. J., at a cost which has been estimated at $5 per ton less than by an all-rail haul east, covering a distance of only 2500 miles.

At Detroit there is annually a rail movement of 25,000,000 tons. About 6,000,000 tons goes to seaboard points. Approximately 80,000,000 tons of freight passes through the Detroit River annully.—W. L. Saunders.

The Group-Bonus Plan

KARL WENNERLUND'S paper on the Group-Bonus Wage Incentive Plan was received with great interest by the men who attended the Production Meeting in Detroit. A number of questions were asked of Mr. Wennerlund at the meeting and answered by him in the discussion of the paper. These questions and answers, published in this issue of THE JOURNAL, are a valuable supplement to the paper itself. The effect of the group-bonus plan upon inspection methods is treated; the division of earnings among the workers in the group is explained; recommendations of the proper size of a group are made. Mr. Wennerlund tells what is done when a worker is discharged and how a new and untrained worker is added to the group. Endorsements of factory executives who have successfully applied the group-bonus plan are included in this discussion which will be found on p. 28.

Aeronautical Safety Code

IT is expected that at an early date regulatory legislation on the use of aircraft will be passed. Detailed regulations should not be included in statutes. This is one of the policies in accordance with which the Society and the Bureau of Standards, which are sponsors for the Sectional Committee on Aeronautical Safety Code, are proceeding. The customs and practices in aeronautics will require modification and change from time to time to keep pace with rapid progress of the industry. If no appropriate engineering standards exist at the time Governmental agencies are authorized to promulgate legal regulations, the latter will have to prepare something of the kind themselves. The Sectional Committee is endeavoring to have ready for use a safety code prior to the initiation of legal regulations, so that these can be based on the results of cooperative technical deliberation. Parts of the safety code have been outlined in a preliminary way. In this work over 200 persons other than members of the Committee have been consulted. It is the view of Chairman Crane that the Committee should consider carefully every matter of engineering importance in aviation, from the design and fabrication of the

aircraft to its piloting, and endeavor to formulate a sound recommendation with regard to it.

Cooperative Fuel-Tests

THE cooperative motor-fuel investigations of the American Petroleum Institute, the National Automobile Chamber of Commerce and the Society of Automotive Engineers, which have already borne results of great value, were made possible by the long, arduous and persistent study and advocacy of a few engineers and scientists who saw clearly what should be attempted and very generously devoted much of their knowledge and time for the common good. They made and revised again and again written statements of the salient points to be considered, and specified how these points should be studied, and how and at what cost the work to this end could be carried on. After many months of work the actual tests were started, splendid cooperation being given by the Bureau of Standards, and by many oil-refining and motor-vehicle producing companies. The Bureau of Standards had staunchly supported and aided the plan from the start, proceeding step by step with the Research Departments of the American Petroleum Institute and the Society of Automotive Engineers. The Bureau of Mines also has rendered valuable assistance.

Automotive Engineering Progress

THE publication of eight Annual Meeting papers in this issue of THE JOURNAL reflects active study being given automotive problems by those engaged in the engineering branch of our industry. They are representative of the manner in which automotive problems are being attacked today. It seems only a few years ago that automotive engineers were accused of applying cut-and-try methods in most of their work. Surely, careful reading of the papers of Messrs. Heron, Young, Kegerreis, Pulsifer and the others will convince one that scientific analysis has found its place among automotive men. The report of the cooperative research work on fuel-volatility should receive the fullest consideration of every engineer employing internal-combustion engines of the light-fuel type. The conclusions set forth have a bearing on the economics of our own industry and that of the petroleum refiners. Truly, this issue of THE JOURNAL records automotive engineering progress of the most creditable type and is deserving of more than average study.

Accidents from Another Viewpoint

THE annual toll of life due to accidents involving motor vehicles is a constant source of anxiety to the industry as well as to the public at large. The recent census returns giving the figures for deaths due to motor-vehicle accidents during the year 1921 has been the subject of much critical comment in trade journals and in the daily press. A perusal of these rather startling statistics raises the question as to their significance. Accidents are obviously dependent upon the density of traffic as well as on density of population.

The figure quoted for 1921, 10.13 fatalities per 100,000 of population as against that for 1917, 7.96 per 100,000 of population, shows an increase of 28 per cent in the death rate per 100,000 population. These figures, however, take account of the increase in population, but not of the increase in traffic.

It is necessary to look elsewhere for figures showing the density of traffic. In 1917 the total number of motor vehicles registered throughout the Country was 4,992,152, or practically 5,000,000. In 1921, the number was 10,505,660, an increase of 110 per cent. In other words, the density of traffic has increased more than three times as fast as the number of fatalities, or, expressed in another way, in 1917, 5,000,000 vehicles were responsible for 8,435 fatalities, and in 1921, 10,500,000 vehicles were responsible for 11,953 fatalities. In 1917, therefore, there were 169 fatalities per 100,000 vehicles as against 114 fatalities per 100,000 vehicles in 1921. In other words, the fatalities per 100,000 vehicles have decreased by roughly 33 per cent.

Commercial Airplane Design Fundamentals

WHAT characteristics must a satisfactory commercial airplane have to assure its operators an adequate return on their investment? Until this question is answered, commercial aviation will make little progress in this Country. Most engineers agree that the Air-Mail Service presents the most fertile field for the airplane at this time. Transportation of package freight is the next step and passenger carrying lies beyond. This seems to dictate the confining of design effort to mail and freight-carrying craft and a discussion of their basic features. Some agreement regarding the principal design and performance characteristics of these machines must be reached among aeronautical engineers before their efforts can be productive of a practical and economic design. Details may be varied to suit the whims of individuals but certainly there must be some fundamentals on which all should agree. The Society has arranged an aeronautical session during the Annual Meeting for the specific purpose of attempting to reach such an agreement. Those members who look upon the commercial application of the airplane in a serious light should avail themselves of this opportunity to express their opinions and foster, thereby, a common-sense attack upon the engineering and economic problems retarding commercial aviation.

Are Section Dues Justified?

SOME members of the Society object to paying dues to local Sections, feeling that their national dues entitle them to participate in Section activities and that the local dues are not justifiable as an added levy. A few minutes' study of the facts in the case should convince one that this attitude is incorrect. There are approximately 5300 members in the Society; only 3000 of these reside within commuting distance of the centers where there are organized Sections. The other members cannot enjoy Section privileges because of their geographical location.

The Society expends $2.95 per member on the Sections annually, $6,500 of the total Sections' budget for this fiscal year being in the form of direct appropriations to the several Section treasuries. This expenditure of a portion of the dues of members who do not reside near enough to headquarters of Sections to join them is justified only by the fact that the Sections produce valuable papers for THE JOURNAL which can be read and enjoyed by such members. Further expenditure of their dues for Sections' operations is, it is believed, not justified; the balance of the expense should be borne by members of the Sections who enjoy all the advantages of membership in them. They alone can attend the meetings, dinners and social functions and this service is worth the nominal sum charged by the Sections as dues. It seems clear that the total expense of the Sections cannot be levied equitably upon the national dues of all members regardless of their location. The proper solution appears to be to expect those who receive Sections service to pay something additional for it.

Society's Research Sheds New Light on Fuel Problem

Eng. Abstract.

By V. H. GOTTSCHALK[1]

ANNUAL MEETING PAPER *Illustrated with* CHARTS

THE author describes a series of road service-tests, made of stock cars driven by their usual drivers when using fuel of specified grades, to determine the effect of any changes in the fuel volatility on the gasoline mileage for the respective make of car as part of a general research program undertaken jointly by the automotive and the petroleum industries. The object was to determine the best fuel as regards volatility, from the general economic standpoint, and what grade of fuel will afford the maximum car mileage per barrel of crude oil consumed in its production.

Factors influencing the selection of cars used are enumerated and the fuels tested are discussed, together with general comment and a description of the test procedure. The results are tabulated and commented upon at some length, inclusive of descriptions of the methods. A summary of the results is presented in the form of conclusions that are stated in four specific divisions. The value of statistical research is illustrated by the results obtained.

SOME months ago, a number of automobile producing companies, at the instance of the Society's Research Department, undertook a series of road service tests to determine the effect of any changes in the fuel volatility on the gasoline mileage for their respective makes of car. These tests were instituted as part of a general research program undertaken jointly by the automotive and the petroleum industries to find an answer to the question, "What is the best motor-fuel, as regards volatility, when considered from the point of view of the general economic situation?" Perhaps the most important phase of this question is, "What grade of fuel will afford the maximum mileage per barrel of crude oil consumed in its production?" The tests here described were intended to answer this question with reference to a number of makes of passenger car when used in normal service under the general supervision of the experimental departments of their respective builders.

To make the results as nearly comparable as possible, and to furnish a central office for the collection and correlation of the results, the following general plan of procedure was formulated by the Research Department in consultation with a number of companies. While each of the individual laboratories was at liberty to modify the plan as desired, as a matter of fact, the original schedule was followed closely in nearly all instances.

The general plan was to operate as large a number of stock cars as practicable successively on each of four specified grades of fuel, the cars to be driven by their usual drivers, in most cases owner-drivers, in the regular course of service, and to secure accurate records of the fuel-consumption, the mileage and the oil consumption and of the crankcase-oil dilution when practicable.

The question of carbureter adjustment, which is obviously of primary importance, had to be handled differently in different cases because of the divergence of

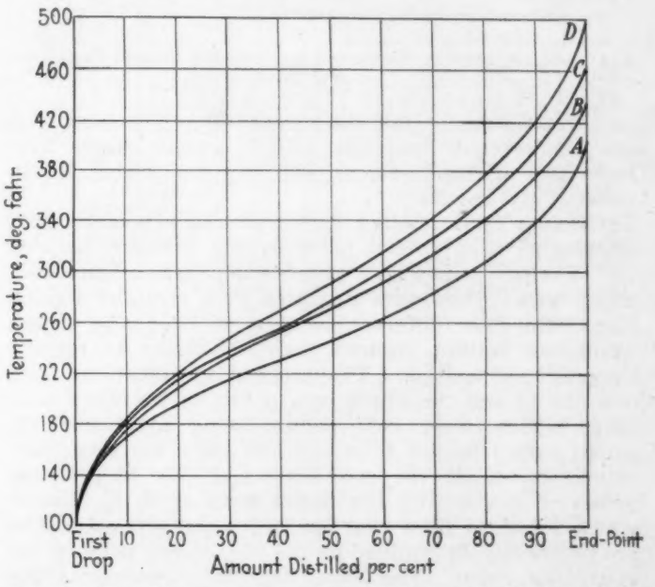

FIG. 1—DISTILLATION CURVES OF THE FOUR FUELS USED IN TESTS

the types of carbureter equipment in the various makes of car. Records of the oil consumption and the crankcase dilution were made by some companies and not by others.

The general conduct of the tests as carried out with only minor variations by all of the 10 companies was as follows:

CARS

Cars were selected for the tests in groups of four so far as possible. The reason for this arrangement in fours will appear later. These cars were, for the most part, owned and driven by officials of the companies, although in a few cases they were test cars or service cars belonging to the companies. They were selected with a view to being typical of the cars of that particular model in actual use throughout the Country. Therefore, they were cars that had received no special overhaul or preparation, except in a few cases where their condition demanded it before starting the tests. These tests involved 4 weeks of operation without any special attention or adjustment, except as to the carbureter for using different fuels.

The number of cars selected by the different companies varied from 4 to 12, and represented from one to three different models.

FUELS

In the selection of fuels of different volatilities for these tests several of the oil companies were consulted, and it was finally decided to specify four grades of fuel as nearly similar as possible except as to the volatility of the heavier fractions, but covering a volatility range as wide as was considered practicable for use in standard

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RECORD OF CARS IN THE TESTS

Car No.	Age	Total Mileage	When Last Overhauled	Carbureter*	Remarks
1					
2					
3					
4					
5					
6					
7					
8					

Please state whether or not carbureter setting was changed for the different fuels and, if so, the method used for making the settings

* *Note make and model of carbureter*

FIG. 2—FORM USED IN RECORDING PARTICULARS OF THE CARS USED IN THE TEST

carbureter and manifold equipment. It will be seen from the results that even less volatile grades might have been used without serious difficulty and added to the value of the results.

The four fuels selected were produced in quantity and distributed at a special price by the Sinclair Refining Co. The curves in Fig. 1 give the distillation figures for these fuels. They were produced by a straight distillation from four different batches of the same crude, which, we believe, insured their similarity as regards chemical composition. The original specifications fixed only the 10 and the 90-per cent points of the distillation curve. The 10-per cent points, being practically the same, were intended to insure the same starting characteristics, while the specification of the 90-per cent points is in line with the recent work of R. E. Wilson, which seems to show that the heavy-end characteristics are practically determined by the 85-per cent point of the distillation curve. The 90-per cent point represents this almost as well as the 85-per cent point and is one of the regular test-points.

These fuels were sent out under labels *A*, *B*, *C* and *D*, in order of the increasing end-point. Careful precautions were taken by all the companies to avoid mistakes in the records involving the grades of fuel. Samples of the four grades were sent to the Bureau of Mines for check-tests, and in most cases the companies' laboratories made their own tests for verification of the labels.

DRIVERS

As stated before, most of the drivers were owners of the cars they drove, but some of the cars were service or test cars driven in accordance with the usual procedure. Provision was made to insure that the drivers should have no information as to the grade of fuel in use at any particular time. This was considered important to avoid prejudice. To what extent this requirement was actually carried out, we have no means of determining, but an examination of the results satisfies us that they are free from any such effect as a preconceived opinion on the part of the driver.

RECORD FOR WEEK BEGINNING..... ENDING.....

CAR NO.	FUEL	MILEAGE, START	MILEAGE, FINISH	MILES COVERED	GAS, START	GAS, FINISH	GAS USED	OIL* USED	OIL SAMPLE MARK	REMARKS

CARS: 1,2,3,4,5,6,7,8 FUELS: A,B,C,D

First Week: 1 and 2 use fuel A; 3 and 4 use fuel B; 5 and 6 use fuel C; 7 and 8 use fuel D
Second Week: 1 and 2 use fuel B; 3 and 4 use fuel C; 5 and 6 use fuel D; 7 and 8 use fuel A
Third Week: 1 and 2 use fuel C; 3 and 4 use fuel D; 5 and 6 use fuel A; 7 and 8 use fuel B
Fourth Week: 1 and 2 use fuel D; 3 and 4 use fuel A; 5 and 6 use fuel B; 7 and 8 use fuel C

* *If lubricating oil is changed for each week's run, record the total amount of apparent consumption by substracting the amount drained from the amount put in.*
If any test is made for dilution, enter results under remarks
Retain samples of new oil and each used oil

FIG. 3—WEEKLY RECORD FORM OF TESTS

TEST PROCEDURE

The test procedure recommended and followed in all cases, with only minor changes, is explained in Figs. 2 and 3. The prime object in carrying out this procedure was to eliminate, so far as possible, the disturbing effects of any differences in the weather and in the types of driving, by a careful selection of a schedule in which they would cancel out, considering the fact that the results derived are merely representative of the differences in mileage for the different fuels. For this reason, a unit of time of 1 week was selected on the general assumption that the normal driving conditions are more likely to be alike from week to week than for any arbitrary period of a different number of days.

The effect of any changes in the weather and of any progressive changes in the condition of the cars was eliminated so far as possible by the schedule of tests. The elimination of systematic errors of this kind is one of the most important features of such a test program.

RESULTS

The nine companies cooperating in the tests used 55 automobiles for 207 separate test-periods of 1 week, averaging 262 miles each. The total mileage, 54,258, was distributed as shown in Table 1, the shortest run being 41 miles and the longest 1111 miles.

TABLE 1—DISTRIBUTION OF MILEAGE IN TEST RUNS

Distance, Miles	Number of Runs
41-99	13
100-199	61
200-299	72
300-399	40
400-499	11
500-599	3
600-699	0
700-799	3
800-899	0
900-999	2
1000-1111	2

The figures for the gasoline mileage, the lubricating-oil mileage and the crankcase dilution are given in Tables 2, 3 and 4.

The second to fifth columns inclusive of Table 2, and Table 3 contain all results reported on the gasoline mileage; the second to fifth columns of Table 2 give the figures arranged in order of the increasing mileage with *B* fuel, which most closely approximates the average filling-station gasoline; Table 3 gives the ton-miles per gallon arranged to show all the group tests, each group consisting of from three to five cars of the same model; the nine cars for which no figures are given in Table 3 include one group of three cars of widely different types, a single ½-ton truck, a seven-passenger phaeton and a commercial car, as well as three closed cars belonging to groups 1 and 9. The last three columns of Table 3 express the individual group-averages as percentages of the corresponding *B* average.

The last four columns of Table 2 record all of the figures reported for the apparent oil-consumption and

are not corrected for the crankcase-oil dilution. Table 4 gives the results on the latter.

Gasoline Mileage

The averages at the bottoms of the second to fifth columns of Table 2 are approximately equal, the maximum difference being only 1.6 per cent. A graphical comparison of the numbers in the column headed *B* with those in the columns headed *A*, *C* and *D* is given in Fig. 4 where the *B* values are measured along the horizontal lines *OO* and the deviations of *A*, *C* and *D* are plotted vertically above or below the corresponding *B* value according as these deviations are positive or negative; that is, in Fig. 4 the abscissas for each one of the three graphs are the *B* values given in the third column of Table 2, while the ordinates are the differences between the *B* value and the corresponding *A*, *C* or *D* values. It will be noticed that the points along the same vertical line cutting all three graphs represent the same car. The method of plotting used for Fig. 4 assumes that the *B* values are correct, and thus distorts the actual variations in *A*, *C* and *D*. In presenting the results in this manner, any variation in the gasoline mileage due to a change from the standard fuel *B* will be accentuated, and it is with the object of facilitating the detection of such variations that this unfavorable representation of the *A*, *C* and *D* runs has been prepared. On this understanding, we may deduce the following facts from Fig. 4:

(1) There is no tendency of the points on any of the three plots to drift from a higher value than *B* at either end of the series to a correspondingly lower value at the other end

(2) The horizontal spacing of the points is not uniform, there being a dearth of points particularly at the higher mileages and to a lesser extent at the lower mileages

(3) At the points *r*, *s*, *u* and *v* of Fig. 4, the *B* values are much higher than the corresponding values for *A*, *C* and *D*

(4) There is a slight, but nevertheless unmistakable, tendency for the *B* values to be the highest of the four

What is meant by item (1) is that an oblique line like *lm* or *pq* does not represent the assemblage of points as well as does the horizontal line, and we may conclude that *A*, *C* and *D* are equivalent to *B* as regards gasoline mileage, not only on the average for all cars, but over the whole range of mileages from 9.5 to 22.5 miles per gal., except possibly at the ends.

The evidence from Fig. 4 supplements the arithmetical equality of the averages for fuels *A*, *B*, *C* and *D*, by demonstrating that the observed mileages for *A*, *C* and *D* oscillate about the corresponding *B* observations practically at random. We are justified therefore in asserting that, within the degree of accuracy attained in the cooperative fuel-test, the gasoline mileage is not dependent on the fuel volatility within the range shown by fuels *A* to *D*, except, again, possibly at the ends of the range.

A part of the condition mentioned in (2) is due to the method of plotting, as for instance the last point, which happens to have a very high *B* value coupled with unusually low *A* and *D* values. Even when the figures are replotted as deviations from the average for each car, the points are more closely packed in the range from 11 to 19 miles per gal. than at the ends. No attention need be paid to this thinning-out of points at the ends, if we may assume that additional runs with cars giving from 8.5 to 11.0 and from 19.0 to 21.5 miles per gal. would yield points which, when plotted, would oscillate about *B* in the same manner as shown for the inner ranges of the graphs; if we wish to be conservative, however, we might qualify the assertion made at the end of the preceding paragraph so as to limit the conclusion to mileages between 11 and 19.

TABLE 2—GASOLINE AND APPARENT LUBRICATING OIL MILEAGES

No. of Car	Miles per Gallon of Gasoline				Apparent Miles per Gallon of Lubricating Oil			
	A	*B*	*C*	*D*	*A*	*B*	*C*	*D*
1	10.73	9.43	12.77	11.62	570	344	1,490	494
2	9.29	9.88	9.02	8.39	...	...	...	...
3	...	10.18	...	...	...	...	...	...
4	...	...	8.54	8.37	...	...	...	...
5	14.07	10.31	12.83	10.85	...	...	...	...
6	8.00	10.63	8.75	13.66	...	...	...	...
7	10.50	10.81	11.39	11.79	...	...	...	...
8	11.21	...	...	...	...	...	...	...
9	12.19	11.52	11.68	12.08	300	287	347	315
10	11.57	12.00	11.57	12.91	...	...	...	...
11	12.57	12.37	9.66	8.76	622	583	418	671
12	12.53	12.45	12.55	12.70	720	265	756	1,720
13	11.19	12.80	12.34	12.62	535	593	520	1,245
14	13.49	12.92	12.68	12.27	1,781	1,344	1,381	1,600
15	9.19	13.13	8.31	7.94	...	...	...	...
16	14.03	13.29	14.40	14.19	...	...	...	...
17	13.05	13.30	14.10	14.08	287	195	576	334
18	13.78	13.35	13.95	12.50	523	1,236	382	768
19	14.58	13.45	13.76	14.62	1,655	...	2,100	754
20	13.70	13.85	15.05	13.15	366	790	872	1,088
21	12.48	13.90	12.28	14.15	...	633	1,218	598
22	...	...	14.12	...	...	...	...	...
23	14.06	14.13	15.48	15.83	...	...	...	...
24	12.87	14.17	14.42	13.59	263	244	232	214
25	14.34	14.38	15.14	14.05	441	412	446	450
26	12.28	14.58	13.70	12.65	400	407	564	504
27	15.86	14.61	15.70	19.98	...	...	...	...
28	13.90	14.82	14.45	15.00	780	637	1,260	1,320
29	15.02	14.98	16.19	14.79	1,352	1,152	475	1,225
30	15.10	15.15	16.50	16.10	826	793	792	1,168
31	16.85	15.35	16.73	15.35	574	570	735	532
32	15.25	15.38	15.00	17.20	319	2,860	...	1,770
33	15.22	15.45	15.65	14.90	530	248	826	1,060
34	16.50	15.50	15.90	15.72	527	584	740	...
35	14.14	15.54	15.83	17.19	386	525	275	352
36	15.70	16.35	16.83	13.75	419	820	731	710
37	16.29	16.35	15.23	16.00	414	533	528	1,944
38	12.50	16.45	12.27	13.45	533	...	1,211	416
39	14.50	16.48	14.40	...	675	(*a*)	1,625	...
40	15.75	16.52	14.92	14.90	...	...	...	...
41	14.06	16.58	14.65	14.52	870	3,290	2,099	1,730
42	16.06	16.59	16.40	16.76	...	...	...	...
43	17.42	16.72	17.60	16.59	832	...	1,408	...
44	16.61	...	17.52	17.37	...	...	...	...
45	18.08	17.02	15.38	16.81	...	...	...	...
46	14.13	17.13	14.72	14.19	520	480	537	517
47	18.47	17.24	18.63	15.54	505	776	484	2,539
48	18.18	17.41	17.08	18.01	...	...	...	...
49	18.86	18.37	17.58	18.79	347	269	313	376
50	20.34	18.48	17.94	17.90	374	389	437	990
51	19.90	18.58	17.10	19.15	1,885	2,195	(*a*)	3,520
52	21.50	19.30	19.68	18.61	872	1,032	(*a*)	99
53	23.58	19.59	19.41	20.58	566	431	526	343
54	22.04	21.00	21.03	21.78	353	988	622	686
55	16.66	22.48	20.13	16.32	400	438	352	(*a*)
Average	14.81	14.94	14.70	14.71	648	768	803	910

(*a*) More oil at the end of the run than at the start.

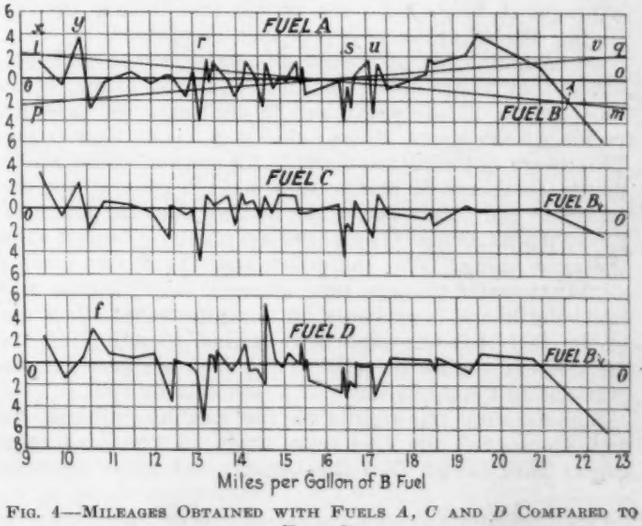

FIG. 4—MILEAGES OBTAINED WITH FUELS *A*, *C* AND *D* COMPARED TO FUEL *B*

The condition mentioned in (3) is partially neutralized by low *B* values at *x* and *y* and by the two isolated high *D* values at *f* and *g*. These exceptional values should not be regarded as being in error, as it is an inevitable consequence of the conditions under which the test was conducted that there are occasional runs showing abnormally good or bad performance; the scheme of using groups of cars was planned to distribute these occurrences among the four fuels evenly.

How this compensation was to be accomplished with a minimum number of test-runs may be understood from the following schedule according to which the cooperating companies agreed to conduct their tests.

	First Week	Second Week	Third Week	Fourth Week
Car 1	Fuel *A*	Fuel *B*	Fuel *C*	Fuel *D*
Car 2	Fuel *B*	Fuel *C*	Fuel *D*	Fuel *A*
Car 3	Fuel *C*	Fuel *D*	Fuel *A*	Fuel *B*
Car 4	Fuel *D*	Fuel *A*	Fuel *B*	Fuel *C*

This scheme provides for eliminating all daily variations by giving a weekly average for each car on each fuel; it compensates for any weekly variations, in the weather for example, by requiring all four fuels to be in use throughout the whole time; it anticipates the effect of any progressive change in the driving properties of each car by preserving a systematic rotation of the fuels for each week. The group average, then, may be regarded as representative of the performance of the particular type of car used in each group.

A closer examination of the abnormal places marked on the graphs, especially when considered in relation to the group runs, indicates that the reservation made in (2) above is of consequence only at the high-mileage end, because the first seven or more points at the low-mileage end represent a heterogeneous set of cars; of various makes, weights and ages, no two of which belong to the same group, and because the abnormal points in the inner range are surrounded by points from the same group, while, on the contrary, the last five points on the high-mileage end all belong to two groups of cars typical of the lightest-weight automobiles. It is possible, as far as may be judged from the mileages reported for cars going 19 or more miles per gal., that additional data on such high-mileage automobiles might reveal a definite, but slight, difference in the mileage with fuels of different volatilities.[2] That this compensation has been realized to a great extent may be seen by examining the averages in Table 3 for the groups that include the cars giving the exceptional points just mentioned.

An examination of the 51 *A* points plotted in Fig. 4 shows that 24 are above the line *OO* and 27 below; of the 50 *C* points, 22 are above and 28 are below; of the 49 *D* points, 23 are above, 25 are below and 1 is on the *OO* line. This bears out conclusion (4). Altogether, there are 70 points above and 80 points below and, in general, the distances to the points below are no less than the distances to the points above. The group averages in Table 3 show the same thing. This indication that the *B* fuel gives a slightly greater mileage than either a more volatile or a less volatile one is probably an uncompensated effect due to the fact that the experience gained in adjusting the carbureters applies directly to the *B*-grade gasoline but does not suit either a more or a less volatile fuel.

Summarizing the results on the gasoline mileage, we find, then, that the fuel tests made by the nine companies have shown that the average car gives approximately the same mileage with the *A*, *B*, *C* or *D* fuels in warm weather.

[2] The Bureau of Standards test-results bear on this question, as three of the four cars used by the Bureau of Standards fit in at the high-mileage end.

Limits of Accuracy of the Fuel Test

Now that it is evident from Fig. 4 that in any column the mileages oscillate about the corresponding figures in the *B* column, we may eliminate the influence of the different cars. We have done this by expressing the group averages of Table 3 so as to represent the number of ton-miles that each car would have run on the same quantity of fuel as was used to get 100 ton-miles with the *B* gasoline. The advantage of recalculating the results in this way is that we now have, for each one of the three fuels, a series of values that oscillate about the same number, very near 100, throughout the whole range. This oscillation of the percentage numbers about a mean, suggests the possibility of assigning a limit of accuracy within which the numbers, when considered as representing independent observations of the same quantity, establish the mean of the series as the best value.

There are a number of ways of expressing this limit of accuracy, but all of them involve the same two factors, namely, the deviations of the separate observation from the mean, and the total number of observations; we have chosen the least optimistic of these indicators, the mean square of the deviations.

The last three columns of Table 3 give the percentages of the ton-mileage secured with fuels *A*, *C* and *D* when compared with the corresponding *B* ton-mileage as 100; under the mean for each column are given the mean deviations of the mean and of a single observation as calculated by the usual formulas.

$$D_s = \vee [S/(n-1)]$$
$$D_m = \vee \ (S \div [n(n-1)])$$

where

D_m = the mean deviation of the mean
D_s = the mean deviation of a single observation
n = the number of observations
S = the sum of the squares of the deviation from the mean

The mean deviations of the mean are ± 1.6, ± 2.3 and ± 2.6; the large discrepancies in the *C* and *D* averages of groups 4 and 5 account for nearly half of these mean deviations.

According to these calculations, we may claim an accuracy of no more than 2.6 per cent either way. That is, while the individual values show extreme variations, up to nearly 40 per cent in two isolated instances, the average of the variations from the group mean are well below 8.0 per cent and the compensation afforded by the group scheme in 12 cases has made the final result, reliable to about ± 2.6 per cent. From this mean deviation we may infer that if *A*, *C* or *D* were more than 2.6 per cent better or worse than the *B* fuel in producing mileage, the cooperative fuel-test as conducted would have detected the difference.

These results furnish a good illustration of the value of statistical research. The 55 cars employed in this test not only embodied various elements of design, but were in all sorts of condition. Some were new; others had been run 30,000 to 45,000 miles. In one set of 12 cars the carbureters were adjusted for each fuel by the same engineer; in another set of 12 cars the carbureters were adjusted at the beginning and not touched thereafter; a number of the companies depended on the drivers to make whatever adjustments were deemed necessary for best performance. Some cars were provided with hot-spots, stoves and similar devices. In a number of cases the week's regular run of a certain car

was changed from its city route to several days' road-work in the country and vice versa. Other variable factors were the weather conditions, possible mistakes in the speedometer and the fuel-volume readings, errors in the transcription of the records, reliability of the measuring instruments and the varying personal element represented by 55 different drivers. The analysis of the results here presented gives evidence that the cooperative fuel-test as conducted was successful in leading to mean values in which the compensation between this host of accidental variations is highly satisfactory.

Summarizing the accuracy calculation, we can say that ± 2.6 per cent may be assigned as an outside limit to the accuracy within which the conclusion concerning the gasoline mileage may be accepted as correct; and this limit is wide enough to include the results with the lightest cars for which the number of observations is not as great as could be wished.

Drivers' Comments

Three of the cooperating companies inserted their drivers' comments on the relative performance of their cars with the separate fuels. The comments of these 25 drivers indicated that greater volatility of the fuel leads to a greater ease and satisfaction in driving, while a high-end point gasoline, such as *D*, induces "spark-knock," slowness in warming-up, sluggish acceleration and other difficulties. One driver considered all four fuels about the same, and in two other instances only did the driver's order of preference not conform to the order of volatility, *A*, *B*, *C*, *D*.

The three companies reporting drivers' comments represented the heaviest and the lightest cars and one of the medium-weight models, and it may be not wholly a coincidence that the drivers of the heavy car were least disturbed about the differences in the fuels, while the drivers of the light car actually complained about the *D* fuel.

Oil Consumption and Crankcase-Oil Dilution

The figures for the apparent miles per gallon of lubricating oil are listed in the last four columns of Table 2. Records of oil consumption are not available for 25 of the 55 cars tested, and in four instances there is an increase rather than a decrease in the volume of oil in the crankcase. Despite the wide range of variation, the

TABLE 3—TON-MILES PER GALLON OF GASOLINE BASED ON A LOAD OF TWO PASSENGERS OF 150 LB. EACH

Group	*A*	*B*	*C*	*D*	Group Averages *A*	*B*	*C*	*D*	Expressed in Terms of *B* as 100 per cent *A*	*C*	*D*
1	24.7	24.8	27.0	26.3							
	24.8	25.2	25.6	24.3							
	21.3	21.7	23.0	23.0	23.6	23.9	25.2	24.5	98.8	105.3	102.4
2	23.7	24.2	23.6	24.5							
	22.4	22.6	24.6	21.5							
	25.6	26.8	27.5	22.4	23.9	24.5	25.2	22.8	97.5	102.8	93.2
3	27.0	25.3	26.0	25.6							
	20.4	22.7	20.1	23.1							
	20.5	20.3	20.5	20.8							
	22.6	21.8	22.8	20.4	22.6	22.5	22.4	22.5	100.3	99.5	100.0
4.	23.2	24.7	22.5	20.9							
	27.9	....	....	....							
	28.9	29.9	28.9	32.2							
	23.0	32.7	20.7	19.8							
	....	....	21.3	20.9	25.8	29.1	23.4	23.5	88.6	80.5	80.8
5	23.2	21.9	23.8	23.4							
	23.2	23.4	25.6	26.2							
	27.4	....	28.8	28.7							
	26.2	24.1	25.9	32.9	25.0	23.1	26.0	27.8	108.2	112.6	120.4
6	19.2	17.2	17.6	16.6							
	21.0	17.5	17.3	18.4							
	19.7	18.8	18.8	19.4							
	14.9	20.1	18.0	14.6	18.7	18.4	17.9	17.2	101.7	97.3	93.5
7	14.6	16.3	16.6	15.6							
	16.2	17.8	18.2	19.7							
	16.2	19.7	16.9	16.3							
	21.6	21.1	20.2	21.6	17.2	18.7	18.0	18.3	92.0	96.2	97.9
8	24.0	24.8	24.5	25.1							
	27.0	25.4	23.0	25.1							
	23.6	24.7	22.3	22.1							
	27.2	26.0	25,6	26.9	25.5	25.2	23.9	24.8	101.2	94.8	98.5
9	20.0	23.6	20.8	20.6							
	....	....	20.2	....							
	21.6	21.8	21.3	24.4	20.8	22.7	20.8	22.5	91.7	91.7	99.2
10	23.4	22.4	21.9	21.2							
	21.6	28.4	21.2	23.2							
	....	17.6	....	....							
	21.2	25.2	23.6	21.8	22.1	23.4	22.2	22.1	94.5	94.9	94.5
11	19.3	22.1	21.3	21.8							
	26.1	25.9	27.8	25.6							
	25.0	25.0	26.2	24.2							
	20.9	19.9	20.2	20.9	22.8	23.2	23.9	23.1	98.3	103.1	99.5
12	29.5	26.3	25.6	25.5							
	26.3	24.5	26.6	22.2							
	23.2	23.4	21.7	23.8							
	24.8	23.8	25.1	23.6	25.9	24.5	24.8	23.8	105.7	101.3	97.2
				Average	22.8	23.3	22.8	22.7	Means 98.2	98.3	98.1
									Mean deviation of the mean ±1.6	±2.3	±2.6
									Mean deviation of single observation ±5.8	±7.9	±8.9

TABLE 4—CRANKCASE-OIL DILUTION

Fuel	A		B		C		D	
Group	Percentage Distilled Off[3]	Viscosity[4]	Percentage Distilled Off[3]	Viscosity[4]	Percentage Distilled Off[3]	Viscosity[4]	Percentage Distilled Off[3]	Viscosity[4]
Set 1								
9	4.5	176	6.5	160	8.0	138	14.0	105
	8.0	138	9.5	124	11.0	116	15.5	93
	8.0	137	10.0	126	14.0	111	22.0	85
	8.0	127	14.5	103	17.0	94	21.0	78
Group Average	7.1	.....	10.1	.....	12.5	.....	18.1	.....
10	.....	.....	9.0	116	.....	.....	.....	.....
	9.0	118	11.5	110	13.0	100	13.0	106
	5.5	140	10.5	115	7.0	146	10.0	120
	8.0	195	7.0	185	16.0	160	17.0	165
Group Average	7.5	.....	9.5	.....	12.0	.....	13.3	.....
12	7.0	130	7.0	129	8.5	128	14.0	103
	7.0	130	7.0	127	10.0	120	13.5	100
	10.5	139	11.5	111	12.5	104	21.0	81
	5.5	145	7.5	128	7.5	128	.....	.....
Group Average	7.5	.....	8.3	.....	11.3	.....	16.1	.....
Average of the three Groups.	7.4	.....	9.3	.....	11.3	.....	16.1	.....
Average of the Groups Recalculated to Percentage Dilution	8.0	.....	10.2	.....	12.7	.....	19.2	.....
Set 2								
1	7.0	202	8.5	196	10.4	173	12.5	154
	6.5	206	8.3	184	11.0	158	11.2	155
	7.5	184	8.5	152	13.4	140	16.0	125
	5.0	233	6.5	208	8.5	206	12.5	149
Group Average	6.5	.....	8.0	.....	10.8	.....	13.1	.....
2	7.5	187	5.5	225	12.0	165	17.7	113
	9.0	179	13.5	144	12.9	132	15.5	125
	4.4	243	8.0	202	10.3	170	10.5	162
	5.2	219	8.5	192	12.5	158	13.0	138
Group Average	6.5	.....	8.9	.....	11.9	.....	14.2	.....
3	6.0	223	7.5	190	9.5	177	.....	.....
	5.5	221	6.3	214	6.5	206	8.7	189
	5.5	247	5.5	233	9.8	178	15.0	128
	7.2	197	8.0	187	10.5	164	13.0	140
Group Average	6.1	.....	6.8	.....	9.4	.....	13.2	.....
Average of the Three Groups	6.4	.....	7.9	.....	10.6	.....	13.2	.....
Average of the Groups Recalculated to Percentage Dilution	6.8	.....	8.6	.....	11.9	.....	15.2	.....
Set 3								
8	7.8	326	5.8	405	10.0	290	19.0	158
	5.0	380	8.0	353	10.5	293	17.5	180
	8.5	335	4.5	439	12.0	259	24.5	123
	7.5	364	7.8	346	11.5	265	27.0	99
Group Average	7.2	.....	6.5	....	11.0	.....	22.0	.....
Average Recalculated to Percentage Dilution	7.8	.....	6.9	.....	12.4	.....	28.2	.....

Original Oil

	Set 1	*Set 2*	*Set 3*
Specific Gravity	0.903	0.917	0.935
Viscosity at 100 Deg. Fahr., Saybolt sec.	201	301	633
Boiling Point, deg fahr.	572	590	545
Flash Point, deg. fahr.	385	382	380
Fire Point, deg. fahr.	450	438	435
Pour Test, deg. fahr.	...	23	15
Demulsibility	...	26	25
Brand of Oil	S. O. Indiana Formula No. 2999 Engine Oil	Medium Heavy Polarine	..

[3] Dilution figures are given as percentage of gasoline distilled from the crankcase oil.
[4] Saybolt viscosity at 100 deg. fahr.

averages for the *A*, *B*, *C* and *D* columns show a progressive decrease in the apparent consumption as the volatility of the fuel decreases. The extent of crankcase dilution cannot, of course, be deduced from the apparent oil-consumption, but this can be obtained by the use of test figures on the oils drained from the crankcases of 28 of the cars at the end of each period.

The percentages of gasoline distilled from the samples of crankcase oil are given in Table 4, which records also the constants of the three original lubricating oils and the Saybolt viscosities of the oils at 100 deg. fahr. These dilution figures may be taken as fairly representative, because, of the 12 groups of cars given in Table 3, seven are included in the table of dilution figures, three each in Sets 1 and 2 and one in Set 3.

The averages for the percentages of gasoline distilled from the crankcase oils are given for these seven groups, which are numbered to conform to Table 3; also, the general averages for the three sets of analyses have been taken and these latter figures are recalculated to show the actual dilution, that is to say, the percentage of gasoline based on the original oil.

In all cases but one, *A* to *B* in Set 3, the dilution increases progressively from *A* to *D*, with a distinct tendency to show a maximum difference between *C* and *D*.

The unfavorable position of fuel *D* as regards crankcase-oil dilution has a vital bearing on the question which the fuel test was expected to answer. This dilution factor assumes an even more important aspect when it is considered that the tests were all run during the summer months, and for this reason the final conclusion has been deferred to await the results which will be obtained from a projected continuation of the test under cold-weather conditions.

SUMMARY

The conclusions drawn from the results of the tests can be summarized as follows:

(1) The ordinary passenger automobile, driven in the summer, shows no difference in the number of miles per gallon that can be obtained with gasolines of volatilities varying within the limits represented by fuels *A* to *D*. This conclusion has been shown to hold to within ± 2.6 per cent

(2) Drivers preferred the *A* and *B* gasolines; they did not object particularly to the *C* gasoline; most of them noted difficulty in driving their cars when using the *D* gasoline

(3) Crankcase-oil dilution increases progressively with a decreasing volatility, and the rate of increase is more marked for the heavier fuels

(4) The high crankcase-oil dilutions, among other things, have indicated the desirability of resuming the tests in the winter months

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NUMBER AND TONNAGE OF MOTORSHIPS

ONE of the most interesting and significant maritime developments since the war is the rapid increase in the number and tonnage of motorships, an increase that seemingly has not been interrupted even by the extreme depression of the shipping industry during the past 2 years. According to the latest statistics available, during the 12 months ended June 30, 1922, in spite of the severe shipping depression, the tonnage of full-powered motorships of 2000 gross tons or over increased 37 per cent, as against a gain of only 4 per cent for steam tonnage exclusive of motorships. Perhaps, it is because of the shipping depression that the Diesel-engined vessel, with its unquestioned economies in operation, is being so rapidly adopted. On June 30, 1921, there were in the merchant fleets of all countries 145 motorships of 2000 gross tons or over, aggregating 692,000 gross tons; on the same date this year there were 186, totaling 946,000 tons.

The United Kingdom continues to lead, with 336,000 tons on June 30, 1922, an increase of 24 per cent. Sweden registered a large percentage gain but continues in third place. Norway has displaced the United States as fourth on the list. Germany, which did not have enough motorship tonnage to appear separately in the list last year, has this year seven Diesel-engined vessels aggregating 32,083 tons. Danzig has two ships aggregating 12,078 tons; one of these is the Zoppot, of 9932 tons, one of the largest tankers in the world.

The following table gives the number, tonnage and distribution of motorships of 2000 gross tons or over on June 30 of each of the years 1920, 1921 and 1922:

	1920		1921		1922	
Countries	Number	Gross Tons	Number	Gross Tons	Number	Gross Tons
United Kingdom	19	108,713	34	217,104	52	335,766
Denmark	16	93,295	21	121,580	22	129,507
Sweden	14	63,951	20	91,681	27	123,753
Norway	19	68,750	21	85,032	27	120,442
United States	22	59,807	28	86,457	30	101,672
Italy	5	24,102	6	26,449	7	36,476
Other	12	43,071	15	63,467	21	98,325
Totals	107	461,689	145	691,770	186	945,941

The above totals are what may be termed seagoing ships. In addition to these, there were 605 vessels under 2000 tons each, aggregating 234,325 tons, which were equipped with Diesel engines. Last year, on the same date, however, there were only 553 vessels of this kind afloat in various parts of the world, and their tonnage totaled 216,110 tons.

The chief engineer of one of the largest British shipyards stated 2 years ago that he saw a promising future for sailing ships with auxiliary Diesel-engine power. This kind of tonnage has increased from 45 ships, totaling 13,000 tons on June 30, 1915, to 777, of 342,530 tons, in 1921, and to 870, of 353,181 tons, in 1922. The United States holds the lead in this kind of tonnage, with 54 ships aggregating 50,957 tons.—*Economic World.*

Annual Report of the Treasurer

THE report of Treasurer C. B. Whittelsey, prepared in connection with the holding of the 1923 Annual Meeting of the Society, is presented herewith. This report covers the fiscal year ended Sept. 30, 1922, and similar figures for the previous fiscal year are included to enable the members to compare the various items readily. The estimated budget for the current fiscal year ending Sept. 30, 1923, is also presented.

	Fiscal Years 1921	1922
The annual audit of Rankin & Co., public accountants, as of Sept. 30, lists the assets of the Society at	$156,183.80	$154,322.23
They consisted of		
Cash	12,155.13	17,619.66
Accounts Receivable	26,806.93	29,544.64
Government and Railroad Bonds	83,194.38	76,513.13
Accrued Interest	909.30	2,284.04
Inventories	19,460.84	13.888.39
Furniture and Fixtures	8,850.08	8,533.08
Deferred Debit Items	4,807.14	5,939.29
	$156,183.80	$154,322.23
These were offset by		
Accounts Payable	$5,107.77	$20,321.18
Deferred Credit Items	2,760.28	2,868.53
Special Reserves	9,398.54	8,527.86
Total Liabilities	$17,266.59	$31,717.57
Unexpended Reserve on Sept. 30, of each year	$138,917.21	[a]$122,604.66

[a]After deducting deficit $15,327.83 and adjustment $984.72.

The income from operation during the fiscal year ended Sept. 30, amounted to	$229,778.90	$235,009.87
This includes		
Initiation Fees	19,265.00	11,470.00
Dues and Subscriptions	76,456.50	77,353.00
Advertising Sales	123,994.54	128,721.20
Miscellaneous Sales	10,062.86	17,465.67
	$229,778.90	$235,009.87
Additional income was received in the form of		
Contributions	7,500.00	8,000.00
Interest	5,567.21	5,052.07
Making a **Total Income** of	$242,846.11	$248,061.94

	1921	1922
Expenditures for the year ended Sept. 30 were	$233,115.01	$263,389.77
For the following purposes		
Publications	51,425.33	46,012.38
Standards	27,960.60	29,275.00
Meetings	10,379.17	10,024.01
Membership Increase	8,614.79	7,483.88
Sections	10,630.48	9,561.06
Research	1,004.98	15,281.96
Employment Service	1,324.77	3,530.96
Cost of Advertising Sales	23,209.19	34,650.14
Cost of Miscellaneous Sales	6,881.31	10,660.21
General Administration	91,684.39	96,910.17
Total Expense was	$233,115.01	$263,389.77
leaving **Unexpended Income** for the fiscal year	9,731.10	
Deficit		15,327.83
On Sept. 30, 1922, the Society had Government and railroad bonds of	Par Value	$78,000.00
	Market Value	76,185.30
	Purchase Value	76,513.13
and accrued interest of		2,284.04

ESTIMATED BUDGET FOR 1923

Income	
Dues and Subscriptions	$75,000.00
Contributions	7,500.00
Interest	5,000.00
Initiations	13,200.00
Advertising Sales	154,779.25
Miscellaneous Sales	13,106.00
Expense	
Publications	$58,250.00
Sections	10,450.00
Research	19,000.00
Employment Service	4,050.00
Standards	27,450.00
Cost of Membership Increase	10,000.00
Cost of Advertising Sales	40,850.00
Cost of Miscellaneous Sales	8,740.25
Meetings—Net Cost	10,825.00
General Expense	78,970.00
	$268,585.25 $268,585.25

It is the wish and endeavor of the officers of the Society to reduce the expenses so far as is in keeping with the Society's best interests, in the hope of showing at least a small amount of income in excess of expenditures.

RESALE VALUE

THE determining factor in the automobile business is based upon the word "service" and is exemplified in resale value. The thing that will determine ultimate production is not the original list-price but the second-hand price. There are 88 member companies in the National Automobile Chamber of Commerce. Twelve of them build about 87 per cent of the cars the 88 companies produce. Not all of the products of those 12 have a high second-hand value. I asked about 500 leading distributors to tell me the second-hand value of the last three models of the 30 leading producers in the industry. I found that they maintain a list of 12 to 15 automobiles which they consider to be of sufficiently high second-hand value to take them in with the hope of a profit in the second sale.

The dealer is handling about 80 per cent of his business on a time basis. Every time he sells an automobile, he takes in one, two or three other automobiles. He discovers very quickly that there is only one chance for him to stay in the business as a dealer, and that is to find out what cars have sufficient value to enable him to sell them a second time at a profit.

A man who starts in business to do one thing consistently day after day and year after year, and sticks to it, not influenced by this or that dealer or sales manager, or this or that new idea, is likely to dominate his field, because he develops the economic situation from the standpoints of manufacture, distribution and service. The only way to undersell another man is to build a better car.—From remarks of E. S. Jordan before the Detroit Section of the Society.

Vol. XII January, 1923 No. 1

Testing Fuels for High-Compression Engines

By Stephen M. Lee[1] and Stanwood W. Sparrow[2]

Annual Meeting Paper *Illustrated with* Charts and Drawings

THIS paper is concerned with certain of the methods used and results obtained in two investigations of fuels for high-compression aviation engines. The fuels in question are benzol and ethyl alcohol, either alone or as blended with gasoline.

The necessity of mixture-ratio runs in fuel investigations is vigorously emphasized. In that the tendencies of a fuel to detonate or preignite limit the conditions under which it may be used, methods of estimating these characteristics are discussed. Also a graphical representation has been made to illustrate the relation between compression pressure, compression-ratio, volumetric efficiency and indicated power in order that, with a given engine, the effect on detonation of changing engine conditions may be estimated.

Results characterizing the performance of the different fuels are mentioned and particular attention is directed to cases where these results are in contrast with current opinion.

TWO investigations, both relating to fuels for use in high-compression aircraft engines, are being conducted by the Bureau of Standards. These differ only in scope; the one authorized by the Bureau of Aeronautics of the Navy Department is primarily concerned with certain specified fuels, while that authorized by the National Advisory Committee for Aeronautics is without such specifications. This paper deals with such of the methods used and results obtained as are believed to be of interest to the members of this Society.

The majority of aircraft engines are now employing as high a compression-ratio as will permit safe operation with aviation gasoline. It has long been appreciated that increased power and economy could be obtained if a higher compression-ratio could be used satisfactorily. Hence arises the interest in fuels whose tendency to detonate is less than that of gasoline and which therefore will permit the use of higher ratios. Among such fuels are benzol and ethyl alcohol, both of which were specified in the investigation authorized by the Bureau of Aeronautics.

Throughout this paper the term compression-ratio is used interchangeably with expansion-ratio because these two ratios are equal in nearly all existing engines although such need not be the case. It should be remembered that it is from the increase in *expansion-ratio* that the increase in power and economy is obtained, while it is the increase in compression pressure resulting from an increase in compression-ratio that necessitates a non-detonating fuel. Although the valve-timing of an engine can be selected so that the effective compression-ratio is less than the expansion-ratio, such an arrangement decreases the power output for a given piston displacement. Consequently this practice has not been generally adopted and, at present, to obtain the advantage of a high expansion-ratio it is necessary to overcome the difficulties accompanying a high compression-ratio.

Any program of fuel investigation should include measurements of the corrosive effect of the fuel on various metals, its freezing-point, its distillation range and its solubility in other fuels with which it may be blended. Discussion in this report, however, is limited to the results of engine tests. These tests were chiefly concerned with the power and fuel-consumption obtainable at various compression-ratios with the different fuel blends. In addition, since the tendencies of a fuel to detonate or preignite determine how high a ratio may be satisfactorily used, every effort was made to estimate these tendencies.

Methods of Test

Much the greater portion of the work was done with a one-cylinder Liberty engine. While the performance of a one-cylinder engine is not identical with that of one cylinder of a multi-cylinder engine of the same dimensions, yet the results obtained form a satisfactory basis for predicting the performance of a multi-cylinder engine with various fuels or various ratios. To interpret the power measurements properly requires that comparisons be made on the basis of the indicated, or the sum of the brake and the friction, horsepower because of the proportionately higher friction-loss of a single-cylinder engine. However, specific fuel-consumption results even upon an indicated-horsepower basis are not applicable directly to multi-cylinder engines. It is probably possible to get lower specific fuel-consumption with a single-cylinder engine than with a multi-cylinder engine because of the absence of the distribution problem. However, on account of pulsations in the intake system the specific fuel-consumption is usually higher, as was the case in this investigation. These pulsations have another disturbing influence on the engine performance in that they cause wide variations in the volumetric efficiency with change in the engine speed. Since both the power developed and the tendency to detonate vary directly with the volumetric efficiency, it can be easily appreciated why care must be exercised in using results obtained with a single-cylinder engine to foretell the performance of a multi-cylinder engine.

It is the practice in this laboratory whenever possible to maintain a constant air-temperature at the entrance to the carbureter. This is accomplished by heating the air to a temperature a few degrees higher than the highest temperature likely to be found in the laboratory. There is ample evidence that, at a constant air-temperature, indicated power varies directly as the absolute or barometric pressure of the air at the entrance to the carbureter. Power measurements can therefore be corrected to a standard barometric pressure on this basis. For careful measurements of detonation it would be desirable to keep this pressure constant and equal to standard atmospheric pressure. All power measurements in

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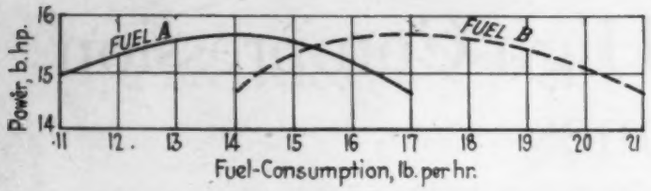

FIG. 1—CURVES SHOWING HOW THE SAME MAXIMUM POWER CAN BE OBTAINED FROM TWO DIFFERENT FUELS BUT NOT WITH THE SAME AMOUNT OF EACH FUEL

these tests were made with an air temperature of 86 deg. fahr. and have been corrected to a barometer reading of 29.9 in. of mercury.

In engine tests of this kind it is necessary that runs be made at many carbureter settings differing by small amounts to determine the relation between, and values of, the power and the specific fuel-consumption for each of the fuels or blends. Such runs were first made with what appeared to be the best spark-advance. To make certain that "appearances were not deceiving," and also

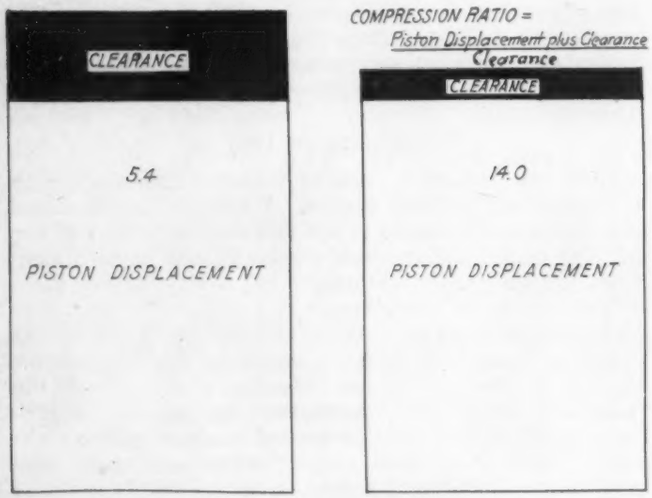

FIG. 2—COMPARISON OF THE COMPRESSION-RATIOS USED IN MAKING THE TESTS

to observe the effect of the spark-advance on detonation, it was customary to make other runs at greater and less advances. The question naturally arises as to whether a single adjustment of the mixture-ratio and the spark-advance would not be sufficient for comparisons of the power, since the power may remain sensibly constant while either the mixture-ratio or the spark-advance is varied a considerable amount. It is true that insofar as determinations of the maximum power are concerned mixture-ratio runs although desirable are not essential. Nevertheless they serve as valuable evidence that correct adjustments were made. However, in fuel-consumption comparisons it is absolutely necessary to make mixture-ratio runs. In some of these tests there was a variation of over 20 per cent in specific fuel-consumption with but 1-per cent variation in the amount of power developed. How uncertain a comparison based on single maximum-power readings would be is easily appreciated. Fig. 1 emphasizes the necessity for mixture-ratio runs in comparing fuels. As will be observed the same maximum-power is obtained with each fuel but not with the same *amount* of each fuel. For example, with a fuel-consumption of 14 lb. per hr., which gives the maximum power for fuel *A*, the engine develops 15.7 b. hp. with fuel *A* or 14.6 b. hp. with fuel *B;* while with a fuel-consumption of 17 lb. per hr. at which the maximum power is obtained from fuel *B*, the engine develops 14.6 b. hp. with fuel *A* or 15.7 h. hp. with fuel *B;* hence a maximum-power setting of either fuel gives a comparison that is unjust to the other fuel. With a carbureter adjustment giving between 11.0 and 15.3 lb. of fuel per hr. more power would be developed with *A* fuel than with *B*. If the adjustment gave a flow of between 15.3 and 21.0 lb. of fuel per hr. the reverse would be true. This sort of condition is likely to result when fuel comparisons are made using fixed sizes of carbureter jet and it is easy to understand why the results of fuel comparisons by different experimenters so often disagree.

It should be noted that accurate measurements of the mixture-ratio are not essential to obtain the benefits of mixture-ratio runs. The results of these and many other engine tests have shown that the quantity of air entering the engine per cycle does not vary appreciably with changes in the mixture-ratio. Hence, for a given engine speed and throttle opening, a curve of power versus fuel-consumption is essentially power versus mixture-ratio, even though the actual mixture-ratio is unknown. This is particularly fortunate, for, while accurate measurements of the fuel-consumption are easily obtained, the situation is very different with regard to measurements of air.

In this work detonation was estimated by sound, by the decrease in power, by the increase in jacket loss and by the destruction of diaphragms so mounted in a spark-plug shell as to constitute an uncooled portion of the combustion-chamber wall.[3] Considerable confidence was placed in the last-named measurement, since it is a gage of the destructive effect of detonation, which phase of the problem is of major importance in aviation work.

Preignition, while easy to recognize, is difficult to measure even comparatively. Mr. Ricardo, in his report on the Influence of Various Fuels on Engine Performance[4], gives self-ignition temperature values for various fuels as determined by conventional methods at atmospheric pressure[5] and by adiabatic compression with air.[6] By the former method the self-ignition temperature of benzol was found to be higher than that of ethyl alcohol, while by the latter method it was found to be lower Which value should be used? Neither method completely simulates engine conditions. This will be evident from what follows. During the compression stroke of an engine there is an increase in the gas temperature due to the heat of compression and a further increase in the temperature of certain portions of the charge caused by the communication of heat from local hot-spots in the combustion-chamber. If the total heat input is sufficient to ignite the charge, the engine will continue to fire when the ignition circuit is broken. So long as ignition by this means does not precede the electric ignition it is unobjectionable. It usually happens, however, that this phenomenon occurs earlier and earlier with each successive stroke until the power delivered by the preigniting cylinder becomes nil. In the case of a multi-cylinder engine one or more cylinders may go into preignition and yet the remaining ones deliver sufficient power to keep the engine running. Upon examination the pistons, valves and spark-plugs in such cylinders will show indications of having been unusually hot. A reasonably satisfactory method for comparing preignition in actual engines is to compare the operation with several spark-

[3] See National Advisory Committee for Aeronautics Technical Note No. 101.
[4] See *Automotive Industries*, May 12, 1921, p. 1003.
[5] See *Automobile Engineer*, May, 1920, p. 199.
[6] See *Automotive Industries*, May 12, 1921, p. 1004.

plugs whose relative tendency to preignite is fairly well known. Suppose that with one fuel a plug that is known to cause preignition readily is used without trouble and with another fuel serious preignition occurs even with a spark-plug that is known to be comparatively free from tendencies to preignition. This is good evidence that the latter fuel is more prone to preignition. Much the same sort of a comparison can be made by noting the difference in the spark-advance that is necessary to produce preignition with different fuels in the same engine. Another method is under development which it is hoped will provide more satisfactory and definite means of measuring these tendencies.

Results

The problem of fuels and compression-ratios can be approached from at least two points of view. One may attempt to select the best fuel-blend for a given compression-ratio or the best compression-ratio for a given fuel-blend. It is along the latter line of endeavor that the most effort has been expended, both in this and in other investigations. The range of compression-ratios studied is indicated in Fig. 2, which shows the volumes before and after compression for the lowest and highest ratios investigated. Results obtained with these and intermediate ratios when using gasoline, benzol and mixtures of the two are summarized in Fig. 3. It is evident that there is an increase in the power and a decrease in the specific fuel-consumption with each increase in the

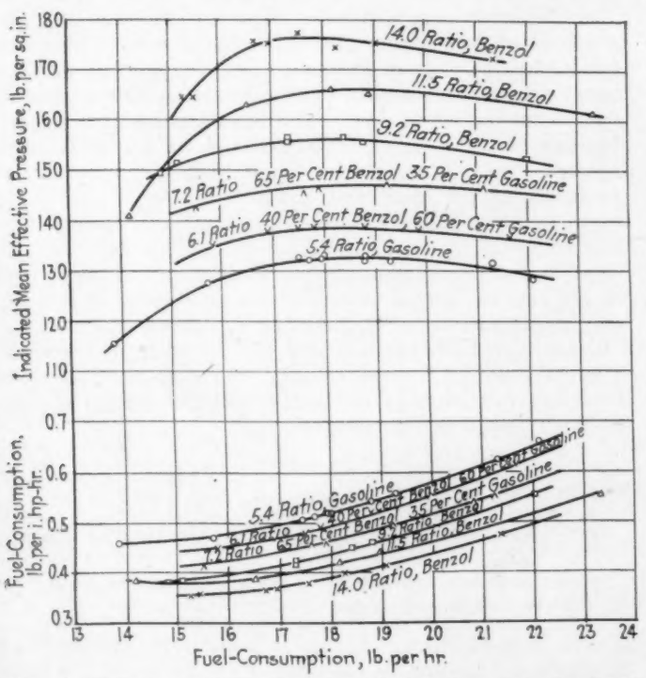
Indicated Mean Effective Pressure, lb. per sq. in.
Fuel-Consumption, lb. per i. hp-hr.
Fuel-Consumption, lb. per hr.
14.0 Ratio, Benzol
11.5 Ratio, Benzol
9.2 Ratio, Benzol
7.2 Ratio 65 Per Cent Benzol 35 Per Cent Gasoline
6.1 Ratio 40 Per Cent Benzol, 60 Per Cent Gasoline
5.4 Ratio Gasoline

Fig. 3—Power and Fuel-Consumption Curves for Different Fuels and Compression-Ratios Obtained from a Single-Cylinder 5 x 7-In. Liberty Engine at a Speed of 1500 R.P.M.

ratio. Since the maximum power is obtained with approximately the same quantity of fuel in each case, it follows that the increase in the power at the higher ratios is not obtained at the cost of a rich mixture acting as an internal cooling medium. The data shown in this figure form the basis for the dotted curve of Fig. 4. The full-line curve of this figure shows the theoretical increase in the power and the efficiency as calculated from the air-cycle efficiency, using the lowest ratio, 5.4, as the basis of comparison. Circles represent actual experimental determinations and it should be noted how close to the theoretical curve they fall. It is interesting to observe how closely the theoretical increase in the power

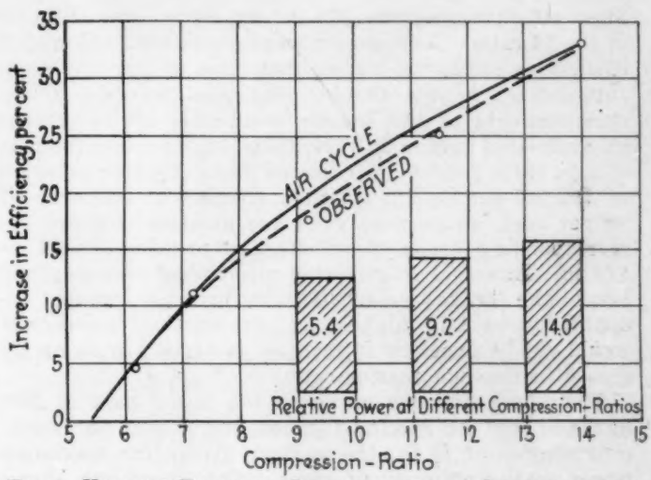

Fig. 4—How the Efficiency Increases with an Increase in the Compression-Ratio

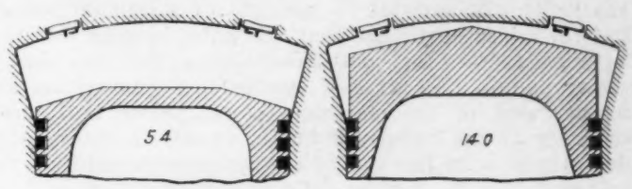

Fig. 5—Combustion-Chamber Shapes Used in the Tests

was approximated even with a combustion-chamber of the shape shown in Fig. 5. Needless to say, this shape was the result of necessity, not choice.

In Fig. 6 is shown the maximum indicated mean effective pressure obtained with each of the various ratios at an engine speed of 1500 r.p.m. The highest indicated

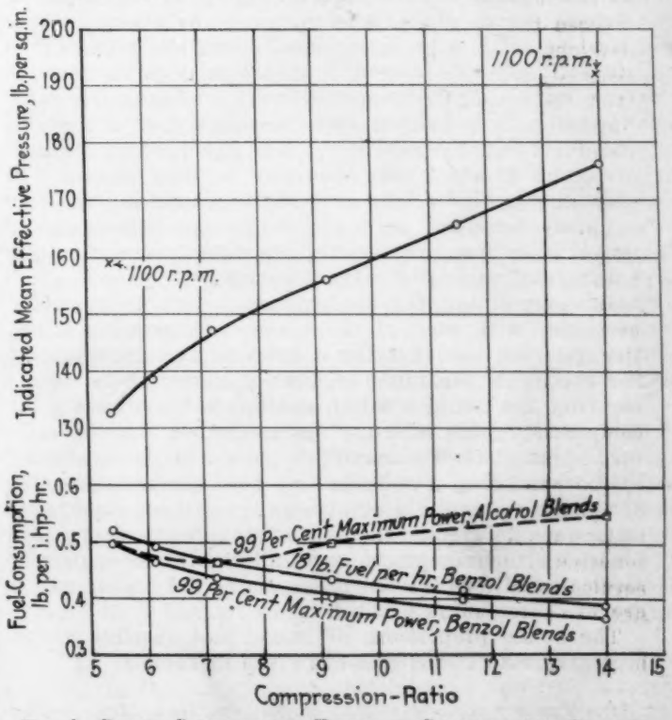
Indicated Mean Effective Pressure, lb. per sq. in.
Fuel-Consumption, lb. per i. hp-hr.
1100 r.p.m.
1100 r.p.m.
99 Per Cent Maximum Power, Alcohol Blends
.78 lb. Fuel per hr. Benzol Blends
99 Per Cent Maximum Power, Benzol Blends
Compression-Ratio

Fig. 6—Curves Showing the Effect of Changing the Compression-Ratio on the Power and the Fuel-Consumption

mean effective pressure, 176 lb. per sq. in., was obtained at the 14 ratio. As some engines operate with volumetric efficiencies somewhat higher than that of this engine at 1500 r.p.m., it was thought desirable to make a few measurements at the higher volumetric efficiency that exists in this engine at 1100 r.p.m. Such measurements at a 14 ratio yielded an indicated mean effective pressure of 192 lb. per sq. in. With a mechanical efficiency of 90 per cent, an average value for aviation engines, the corresponding brake mean effective pressure would be 173 lb. In actual practice the mechanical efficiency and hence the brake mean effective pressure would undoubtedly be still higher, as the friction horsepower would not be expected to increase materially with an increase in the compression-ratio.

The upper full-line curve in the lower part of this figure shows the specific fuel-consumption at an hourly consumption of 18 lb., the mixture giving the maximum power in the majority of cases. The lower line shows the specific fuel-consumption at mixtures sufficiently lean to give 99 per cent of the maximum power. This value of 99 per cent is arbitrarily chosen as representing the best economy obtainable with practically full power development. Moreover, a comparison of the fuel-consumption results on the basis of 99 per cent of maximum power has the added advantage that this point is more definite as regards the fuel-consumption than the maximum-power point. The curve of specific fuel-consumption at 99 per cent of the maximum power shows the same tendency as the full-power curve, namely, a continuous decrease with an increase of the compression-ratio up to and including the highest ratio used in this test.

Very different characteristics are shown in the dotted curve of Fig. 6, which is the result of using at each compression-ratio a blend of gasoline and just sufficient alcohol to prevent serious detonation. With the alcohol blends, as with the benzol blends, there was an increase in the power with an increase in the compression-ratio. However, because of the lower calorific value of alcohol, as the necessary percentage of alcohol in the blend increased the weight of fuel required for a given power development also increased until eventually a point was reached where no further increase in economy resulted from increasing the compression-ratio. Recognizing this limitation, it is evident that the selection of an alcohol blend and compression-ratio will be governed by the kind of service to which the powerplant is to be devoted. In short flights the weight of the powerplant and the fuel is chiefly dependent upon the weight per horsepower of the engine. On long flights the fuel-consumption per horsepower-hour is of major importance.

In determining the percentage of benzol or alcohol necessary with each of the various compression-ratios, the criterion was that the detonation and consequently the maximum pressures should not exceed those resulting from the use of aviation gasoline at the standard 5.4 compression-ratio with the spark-advance and the mixture adjusted for the maximum power. It is significant that, when using a suitable fuel at a compression-ratio of 14, the maximum pressures as verified by diaphragm tests were no greater than those resulting from the detonation ordinarily present with a 5.4 ratio under normal service conditions. Mean pressures at the higher ratios are, of course, much greater.

The actual proportions of benzol and gasoline found necessary with this engine are given in Table 1.

[7] See THE JOURNAL, May, 1922, p. 305, and June, 1922, p. 451; also National Advisory Committee for Aeronautics Technical Note No. 93.

TABLE 1—MIXTURES USED WITH DIFFERENT COMPRESSION-RATIOS

Compression-Ratio	Per Cent by Volume Benzol	Gasoline
5.4	0	100
6.1	30	70
7.2	65	35
9.2	85	15
[a]11.5	100	0
14.0	100	0

[a] Although 100 per cent of benzol was used in the tests at this ratio, undoubtedly a blend containing a very small quantity of gasoline would have been satisfactory.

In using alcohol as an anti-detonation agent, it was found to be approximately twice as effective as benzol at the low compression-ratios where the greater portion of the fuel blend was gasoline. At the higher ratios where the gasoline was the lesser constituent there seemed to be little difference between the effect of benzol and that of alcohol. More experiments are needed to determine whether this is a general relationship or the result of a peculiar combination of circumstances. In this connection the two consequences of adding a considerable proportion of benzol or alcohol to gasoline should be appreciated: (*a*) detonation is decreased because the amount of the detonating substance, gasoline, is decreased, and (*b*) detonation is decreased because the amount of the anti-detonation agent is increased. "Dopes" that are effective in relatively small quantities attain their ends in the last mentioned way.

Undoubtedly some investigators will question the proportions of benzol and gasoline cited above. Their own experience may have shown a need for higher percentages of benzol at certain of these ratios. Statements as to the proportion of any anti-detonation agent necessary for engines of a given compression-ratio are misleading because there is ample evidence that the compression *pressure* rather than the compression-*ratio* forms the most reliable gage of the engine's tendency to detonate and hence of its fuel requirements.[7] It follows that the compression pressure and hence the detonation tendencies of an engine of low compression-ratio and high volumetric efficiency may equal or exceed those of an engine of high compression-ratio and low volumetric efficiency. The influence of volumetric efficiency can be understood from consideration of the factors determining the compression pressure. In general,

$$P_2 = P_1 r^n$$

where

n = 1.25 to 1.40
P_1 = the absolute pressure at the beginning of compression
P_2 = the absolute pressure at the end of compression
r = the compression-ratio

It will be seen that a change in volumetric efficiency or barometric pressure by changing P_1 will change the compression pressure P_2 and that the actual amount of this change will increase with an increase in the compression-ratio.

Fig. 7 has been plotted as a sort of birds-eye view of the possibilities open to the engineer in designing an engine to use fuel of known detonation characteristics. It gives the relation between the compression-ratio, the volumetric efficiency, the indicated mean effective pressure and the compression pressure. In the upper portion of the figure a series of compression-pressure curves has been plotted against the volumetric efficiency. These curves were plotted from the relation mentioned above, assuming P_1 to equal 14.7 lb. per sq. in. at 100 per cent

volumetric efficiency and for lower efficiencies to be lower in direct proportion to the efficiency. In securing the lower group of curves the first step was the plotting of an actual measurement of the indicated mean effective pressure obtained at a 5.4 ratio against an actual measurement of the volumetric efficiency. From this point the curves were derived by making two well justified assumptions: (*a*) that at any compression-ratio the indicated mean effective pressure is directly proportional to the volumetric efficiency, and (*b*) that at any volumetric efficiency the indicated mean effective pressure is proportional to the air-cycle efficiency as determined by the compression-ratio.

An example will show best how these curves may be used. Assume a fuel and an engine design such that with an absolute compression-pressure of 160 lb. per sq. in. there is an adequate margin of safety against detonation. From the upper curves it is seen that a pressure of 160 lb. per sq. in. would be obtained with a volumetric efficiency of 71.5 per cent at a compression-ratio of 7 and with a volumetric efficiency of 89 per cent at a compression ratio of 6. The lower curves show an indicated mean effective pressure of 125 lb. per sq. in. to be developed with a volumetric efficiency of 71.5 per cent at a compression-ratio of 7, while an indicated mean effective pressure of 147 lb. is obtained with 89-per cent volumetric efficiency at a ratio of 6. For the 7 ratio the air-cycle efficiency is 54.1 per cent; and for the 6 ratio, 51.1 per cent. Hence, with the lower ratio the indicated power would be 17.5 per cent greater with an efficiency, on the basis of indicated power, 5.8 per cent less.

[8] See *Automotive Industries*, May 12, 1921, p. 1006; also National Advisory Committee for Aeronautics Technical Note No. 93, p. 7.

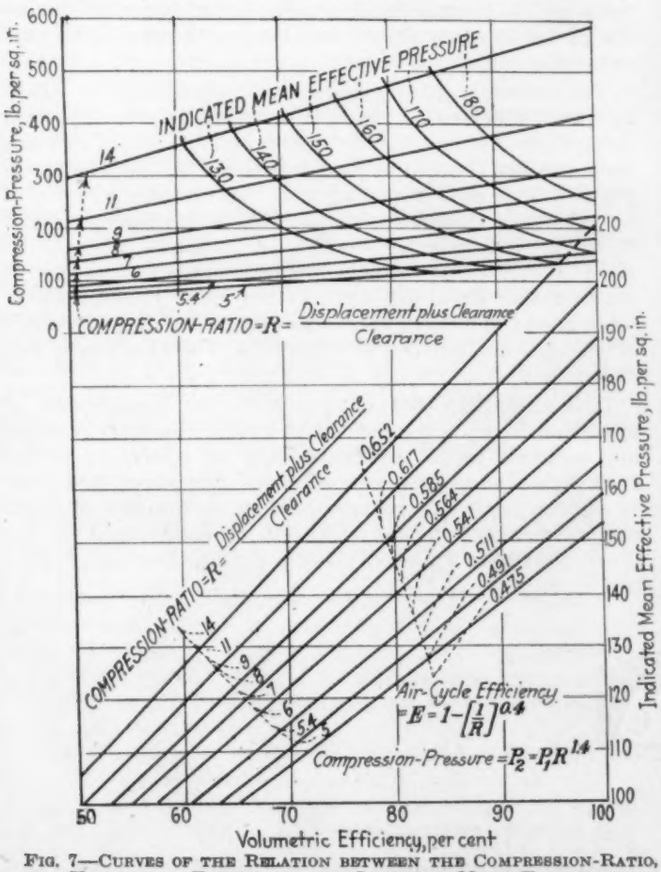

FIG. 7—CURVES OF THE RELATION BETWEEN THE COMPRESSION-RATIO, THE VOLUMETRIC EFFICIENCY, THE INDICATED MEAN EFFECTIVE AND THE COMPRESSION PRESSURES

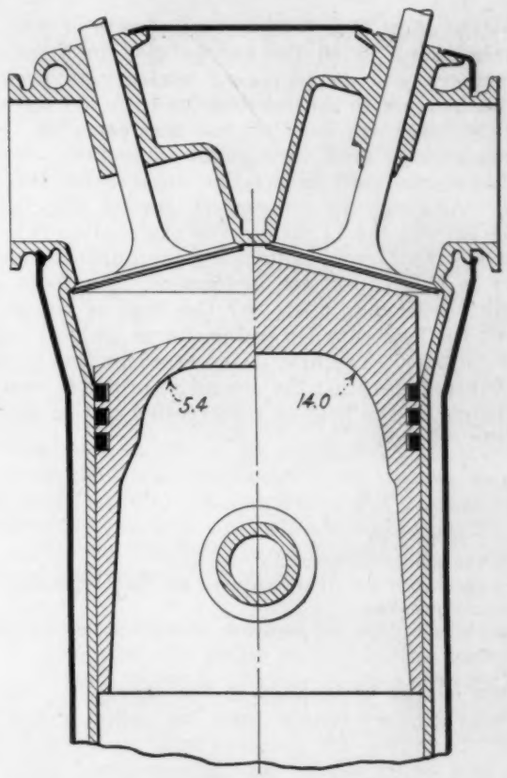

FIG. 8—A COMPARISON OF THE 5.4 AND 14 COMPRESSION-RATIO PISTONS

The engineer who has been confronted with problems of this nature realizes that these curves tell only a part of the story. The engine with the lower volumetric efficiency would need to be larger and heavier to develop the same power. Its mechanical efficiency probably would be low because of its low power per cubic inch of piston displacement. Hence, the difference in brake horsepower for the conditions mentioned would be even greater than the difference in indicated power, while the advantages of the higher ratio from a thermal efficiency standpoint would be somewhat offset by the increased weight per horsepower and the decreased mechanical efficiency. From the foregoing it should not be inferred that the compression-ratio and the volumetric efficiency are the only conditions governing detonation. The size and shape of the combustion-chamber, the location and number of spark-plugs, the cylinder and piston construction all have influences which can be determined only by experiment. Moreover, in maintaining a constant margin of safety from detonation, the permissible compression-pressure does increase slightly with an increase in the compression-ratio. This results from a slight decrease in the ratio of the explosion pressure to the compression pressure.[8]

In spite of these many qualifications this plot, Fig. 7, does make clear why with a given fuel an engine of high volumetric efficiency is limited to a lower ratio than one of lower volumetric efficiency. It shows also that, so long as detonation is the limiting factor, the greatest power output with a given fuel and engine may be obtained by first securing the highest possible volumetric efficiency and then employing the highest compression-ratio that will provide adequate safety from detonation.

In some instances preignition rather than detonation may be the factor that determines the proportions of the blended fuel. This may be the case with alcohol-gasoline blends, for, although alcohol seems to possess more anti-

detonation effect than benzol, it appears to be more prone to preignition. With the engine used in these experiments there was an increased tendency to preignition with an increase in the compression-ratio but the amount of this increase was less than had been expected. In fact, the spark-plugs used throughout these tests were of a type commonly used in aviation engines for the past 4 years. Although in no respect special they gave no trouble even at the 14 ratio. The explanation is not difficult when the factors influencing preignition, mentioned earlier in this paper, are understood. As there stated, preignition depends upon (*a*) the heat of compression, and (*b*) the heat supplied to the charge by local hot-spots in the cylinder. The first is increased by an increase in the compression-ratio; the second is probably decreased. The change in the heat of compression can be estimated from the relation

where

$$T_2 = T_1 r^{n-1}$$

n = 1.25 to 1.40
r = the compression-ratio
T_1 = the absolute temperature at the beginning of compression
T_2 = the absolute temperature at the end of compression

How much heat is supplied to the charge by local hot-spots depends very largely upon the piston and cylinder design. That this will decrease with an increase in the ratio could be shown by comparative plots, based upon theoretical considerations, of the change of temperatures throughout the cycle. A familiar, though perhaps unappreciated, illustration of this same effect is the common expedient of retarding the spark to make an engine heat-up quickly. Obviously, the increased heat is the result of the decreased expansion of the gases after burning. These observations make it clear that engines may require different fuel-blends solely because differences in design give them unequal tendencies to cause preignition. Differences in the tendency toward preignition with different compression-ratios may be less than the difference in the tendency toward preignition of different engines at the same ratio. Therefore, one must be cautious in using test-engine results as a basis for selecting fuels for other engines.

Perhaps undue interest attaches to some of the experimental results because of the degree in which they prove or disprove prevalent opinions. For example, there seems to be widespread belief that alcohol and benzol require a greater spark-advance than gasoline requires. To determine whether there is justification for this belief, runs were made with various spark-advances with each of the three fuels at a compression-ratio and a throttle-opening such that no detonation was present with any of the fuels. However, the maximum power was obtained with the same spark-advance with each fuel. It seems probable that whenever the best spark-advance for alcohol or benzol has been found to be greater than that for gasoline, this fact can be considered an indication that detonation was present when gasoline was used and that for this reason the spark-advance was not used which, in the absence of detonation, would have given the maximum power.

A similar situation exists with regard to the jacket loss, an important item in aviation work since it determines the size of radiator necessary. Some investigators have found a marked decrease in the jacket loss upon changing from gasoline to alcohol. No such effect, however, was found either in these experiments or in experiments with multi-cylinder engines in the altitude chamber. Again the difference probably can be charged to detonation or preignition, whose presence when using gasoline as a fuel caused unusually high jacket-losses which disappeared when these conditions were eliminated by the use of alcohol or benzol.

Another illustration is available of the influence that engine characteristics may have on work of this kind. It will be remembered that with benzol some experimenters have encountered preignition without audible detonation at ratios of less than 8. This might lead one to suppose that some quality of the fuel made detonation impossible. That such is not the case was shown by these experiments, for both alcohol and benzol detonated at the 14 ratio, provided the spark was advanced sufficiently.

To obtain within a reasonable time a piston that would give a 14 compression-ratio the rather crude expedient of welding additional metal to the head of a 7.2-ratio piston was adopted. As shown in Fig. 8, the result was an abnormally thick piston-head. This thick head may have been a real advantage in that it provided a large mass of metal capable of rapidly absorbing heat during the period of combustion when the gas temperatures were extremely high.

The necessity of a proper combustion-chamber shape has been stressed so much that added interest attaches to the fact that very nearly the theoretical power-increase was attained with the unconventional shape shown in Fig. 5. This does not imply that this shape, necessitated by the flared cylinder construction, is desirable, or that it might not give very poor performance at reduced throttle. Even with this shape, however, less spark-advance was necessary than at the lower ratios. Maximum power at the 5.4 ratio was obtained with an advance of 26 deg., while with the 14 ratio a 13-deg. advance was sufficient.

It is axiomatic that knowledge of the possibilities of operation under a certain set of conditions must precede any estimate as to the desirability of operating under such conditions. The sole effort of this paper has been to outline some of the possibilities in the use of high compression and to call attention to some items of interest that have appeared in this explorational investigation.

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Automotive Work as a "Profession"

By Dr. C. A. Prosser[1] Minneapolis Section Paper

THE author outlines what constitutes a "profession" and discusses the factors necessary in the making of a real profession of automotive work. The essentials are stated to be a body of knowledge built up through experience for use as a guide; its collation and organization so that it can be taught to others; the establishment of training schools; recognition by society in general of the social benefit of a profession and consequent financial assistance; the development of a professional consciousness on the part of individuals engaged in the same lines of effort; and a vision, on the part of the individuals engaged in a given line of work, of the social service that they do by the proper pursuit of their own calling.

NO industry in the history of the world has grown so rapidly as that of automotive production. The value of the mechanisms handled, the problems involved in the standardizing of parts, the necessity in the stress of competition for constant improvement in the product and the many applications of scientific principles that the automobile requires, have made necessary the employment of large numbers of technicians and engineers. This technical service concerns itself at the factory with experiments and improvements in the manufacture of the automobile itself. At distributing points throughout the Country, it is more and more concerned with the many problems involved in the repair and sales end of the industry. These problems range all the way from the better organization and administration of the plant and the office to advancements in the processes of repair.

The men who come into this work locally range all the way from graduates in mechanical and electrical courses of engineering colleges, particularly the former, to men who have grown up in the business. They largely combine with a general education and reading a wide experience either in the repair or the sales end of the business. What I have to say has to do with the opportunity and necessity of the local automotive engineer making a real profession of his business.

There are certain earmarks of any work in the world that raise it to the opportunity, spirit and dignity of a profession. I will set them up one by one as a straightedge and point out the extent to which the work in the automotive industry conforms to the same tests.

What Constitutes a "Profession"?

We never have a "profession" until a body of knowledge is built up through experience to guide those who are engaged in its pursuit. This is true of law, medicine, education and all forms of engineering. While the automobile has been in existence only a comparatively short time, science and experience, working together, have built up a rich body of knowledge that a man must have if he is to become a professional servant and leader in the business.

Until this body of knowledge is collated and organized so that it can be taught to others, it does not constitute the basis of a profession. Special courses in automotive and tractor engineering and other practical courses are being established in industrial and trade schools everywhere for the conferring of the knowledge that is being organized into textbook material and published broadcast.

In the truest sense of the word, we do not have a profession until schools are established for training persons who are to pursue it. This step already has been taken. Preparation for the profession of the automotive engineer will, in my opinion, come through the engineering school, where special courses will be established for those who intend to specialize in automotive engineering; through the industrial and trade-school courses, particularly in part-time and evening work; and through correspondence-school courses and related reading on the part of the ambitious man in the trade who has successful experience and seeks to improve his professional standing.

A profession does not amount to much until society comes to recognize the social benefit of the profession and establishes, at public expense, means whereby persons can become better trained for its successful pursuit. As in every other line, history has repeated itself in the case of training for automotive and tractor service. The work begun with private schools is now being taken over rapidly by the public as the social worth of the service becomes recognized.

Professional Consciousness

Until those who are engaged in a given line of work come to develop a professional consciousness, their work is not a profession. This has been true in law, in medicine, in education, in theology and in all the forms of engineering. A man develops a professional consciousness when he comes to have a deep sense of interest and pride in his calling; when he is an earnest student of his line of work; when he is sensitive to the difficulties and weaknesses of his own service and that of his fellows; when he throws himself, heart and soul, into his task, not only improving his own efficiency but advancing the interests of the profession in which he is enlisted.

The professional consciousness must rise to a point such that those engaged in the work associate themselves in local, state or national societies in the interest of their business, whatever it may be, before a profession is constituted. In fact, the man who does not belong to the organized associations in his own line may, in most cases, be rated as one of the less desirable members of his profession.

The professional consciousness of a group of men develops a professional code of ethics or conduct in the relationship of these men with each other, before any line of work becomes a profession. Members need to be, and usually are, jealous of the reputation of the group. They gradually come to establish standards of professional conduct in the performance of work. They do not want the inefficient or the dishonest man to be associated with them; whenever they can, they proceed to drive them out of the business. They work for recognized legal or trade standards of preparation to which they wish those who enter the business or who become associated with them in a society to conform.

As this professional consciousness increases, members

[1] Director, the William Hood Dunwoody Industrial Institute, Minneapolis.

of a group came to have a code of conduct in their relationship with each other. Some of the items in this code of conduct are friendliness with each other in spite of competition, fairness in their dealings with each other, the absence of personal criticism, denunciation and innuendo, and a bond of brotherhood in a common cause. I believe this result will come about in automotive engineering through the association of the members of the Society and with the growth of its professional spirit that is sure to come with the development of the automobile business, the deepening of individual experience and the strengthening of the relationships of the members with each other.

Social Worth and Service

To constitute a profession in the truest sense, those engaged in it must come to have a vision of the social worth and service that they are performing through the proper pursuit of their own calling. One very strong illustration of that at present is the extent to which doctors are giving of themselves in so many ways to clinics, to campaigns for public health, to the fight against disease and to the promotion of preventive measures of all kinds. By the standardizing of their service and the spread of information and procedure, which reduces the amount of sickness and disease, it would seem at first glance as if the doctors were interfering with their own business, because the policy that they are pursuing so earnestly in so many quarters is reducing the number of patients. The doctors have taken the higher ground in this as they have become professionally conscious of the responsibility of their greater knowledge and skill for public well-being. In the end it will all redound to the best benefit of their own profession.

The automotive engineer likewise needs to catch a vision of what efficiency in every phase of the automobile business means to the public in reducing expenditures, in the prevention of accidents, in comfort, in the greater mobility of the human race, in quicker and safer communication and in all the things about the automobile that make it such a marvelous instrument for personal pleasure and social good. Every step that is taken in the improvement of any phase of the automobile makes in this direction, and gives, or should give, the automotive engineer the very great gratification of knowing that the medium by which he makes his living is very much worthwhile socially.

The foregoing is a statement of goals. The preliminary steps have been taken by the Society of Automotive Engineers, and it rests with the local sections of the Society and their individual members whether these ideals shall be realized fully.

DECEMBER COUNCIL MEETING

THE meeting of the Council held on Dec. 5 at New York City was attended by President Bachman, First Vice-President Whitbeck, Second Vice-Presidents Clark, Brautigam and Segner, Councilors Scott and Crane, Past-President Beecroft, Treasurer Whittelsey and A. J. Scaife, nominee for the 1923 Council.

The financial report as of Oct. 31 showed a net balance of assets over liabilities of the Society of $121,806.46, this being $16,039.92 less than the corresponding figure on the same day of 1921. The income of the Society for the first month of the current fiscal year amounted to $21,840.95. The operating expense during the same period was $22,534.30.

One hundred and seven applications for individual membership and 42 for student enrollment were approved. The following transfers in grade of membership were approved: From Associate to Member, O. W. L. Coffin, Herman A. Zeller, R. O. Butterfield, Harvey E. Clay, L. H. Gaylord, H. Borreson, H. G. Prodoehl; Junior to Member, E. L. Apor; Junior to Associate, Emil F. Anderson, M. O. Hodgins, Frank R. Wood, H. Woodall, Scott F. Hunt, Sterling O. Wilson, Tsao Yu.

It was reported that during the month of November 105 applications for individual membership and for student enrollment had been received as compared with 66 applications received during the same month of 1921, and 113 during the same month of 1920. On Nov. 30 there were 5321 names on the rolls of the Society, including affiliate member representatives and enrolled students, as compared with 5246 on the same day of 1921 and 5230 on the same day of 1920.

The following subjects were assigned to various Divisions of the Standards Committee:

Storage Battery Tray Terminals — Electric Vehicle Division
Motorboat Lighting Equipment—Lighting Division
Tail-Lamp Illumination—Lighting Division
Brake Ratchets—Parts and Fittings Division
Flexible Discs—Parts and Fittings Division
Hardwood Lumber—Passenger-Car Body Division
Paints, Varnishes and Enamels—Passenger-Car Body Division
Aluminum Sheets—Passenger-Car Body Division
Footman Loops—Passenger-Car Body Division

Earle Buckingham, E. H. Ehrman and George S. Case were appointed a committee on editorial methods of expressing limits and tolerances in connection with the issuance of data sheets of the Society.

The concluding sessions of the present Council were scheduled for Jan. 8 and 9.

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Automotive Transportation by Trailers

By E. R. Greer[1]

Minneapolis Section Paper *Illustrated with* Photograph

THE author states that trailer operation is a highly specialized form of transportation and has features of flexibility that make it of far greater importance than is generally supposed. Certain specified limitations must be adhered to, because of existing laws and regulations, and there are other limitations due to the laws of mechanics and the present state of development of this form of transport.

The legal restrictions in regard to gross weight are enumerated and commented upon, as well as those applying to wheel loads and to front and rear-axle loads. The cost of operation is divided into seven parts that are specified. In regard to passenger transportation, the suggestion is made that semi-trailers with passenger bodies be used, and six advantages to be gained thereby are given.

THE general public and probably many engineers are inclined to look upon trailers as suitable for use only in special cases. Where roads are good and large quantities of any commodity are to be transported, the idea of using a trailer may occur to anyone; but trailer operation is a highly specialized form of transportation and has features of flexibility that make it of far greater importance than one might suppose. Certain set limitations must be adhered to, due to existing laws and regulations, and there are other limitations that result from the laws of mechanics and the present state of development.

We are permitted to have 800 lb. of weight per inch of tire width, 22,500 lb. gross weight on one axle or 28,000 lb. gross weight on one vehicle. We are allowed a speed of 6 to 25 m.p.h., depending on traffic conditions. Vehicles must not be over 8 ft. wide and 80 ft. long, and they must not be too noisy or destructive.

Supposing that the volume of work to be done is large; to arrive at the maximum economy in automotive transport it is necessary, first, to carry the largest possible load each trip; second, to travel as fast as possible; and, third, to reduce the time of loading and unloading to a low minimum.

A certain amount of weight is required to get the traction necessary to move the load, and the weight must be balanced so that all road conditions can be met. There must be enough weight on the steering wheels to provide positive control when a sharp turn must be made on slippery pavements, and when hauling a maximum load up a 10 to 15-per cent grade. The problem is simple, but it is not often given consideration.

Tires 14 in. wide or a total of 28 in. of tire width per axle will carry 22,400 lb. at 800 lb. per in. of tire width. This is the limit; so, it is necessary to use more axles to carry heavier loads. The logical procedure is to use trailers. A consideration of speed and the avoidance of delays will show at once that a single trailer is enough, even if two trailers could be kept within the 80 ft. of allowable length. A consideration of the balance of the load and economy in operation will show that the two-wheel trailer, or semi-trailer, offers the best possibilities. As a result, it appears that whereas the 6-ton truck represents the limit in truck capacity, the 12-ton semi-trailer is the limit in the six-wheel unit. Such a vehicle is shown in Fig. 1.

Fig. 1—An Application of the Tractor and Semi-Trailer to the Transportation of Merchandise

The 6-ton truck with a 6-ton load weighs about 24,000 lb., with about 20,000 lb. on the rear axle and 4000 lb. in

[1] M.S.A.E.—Vice-President and engineer, Motor Transport Co., Minneapolis, Minn.

front. The 12-ton semi-trailer with a 12-ton load weighs about 28,000 lb., one-third of which is to rest on the truck tractor, that weighs about 7000 lb. The trailer axle carries 18,700 lb.; the rear axle of the tractor, 16,300 lb.; and the front axle, 4000 lb. Hence, the load on the road is lighter in the case of the semi-trailer, although the total net load is doubled.

The 6-ton truck might be hitched to a four-wheel trailer of 6 or even 12-ton capacity, but such a combination is awkward, difficult to operate and expensive. It is a time-waster instead of a time-saver, because the truck itself must be held for loading and unloading, or it could not pull the trailer. It is almost impossible to back up such a trailer. At more than 10 m.p.h. there would be trouble caused by the "snaking" of the trailer, and the wear-and-tear on the truck would be excessive.

The semi-trailer "handles" as a single unit. It can be operated safely at speeds equal to truck speeds. It can be backed and maneuvered in less space than an ordinary truck. It requires less than 2 min. to exchange trailers, thus eliminating all waste of time for loading and unloading. The economy of operation is excellent, and it is a more flexible and adaptable unit for variable loads and conditions. The maximum size of such vehicles has been mentioned, but the same ratio holds good for smaller sizes.

Overloading is general in the case of all trucks; it is a bad practice, but one that is difficult to prevent. When a semi-trailer is overloaded, only one-third of the overload is carried by the tractor; so there is much less chance of trouble from this source.

Cost of Operation

Low cost per unit of work performed is the objective. If all items of cost are considered, the semi-trailer is the most satisfactory in the majority of cases. Costs must include:

(1) General or overhead expense
(2) Driver, helper and compensation insurance
(3) Fuel, lubrication and tires
(4) Maintenance
(5) Overhaul reserve
(6) Substitute vehicle
(7) Fixed costs, consisting of depreciation, interest on investment, garage, insurance, taxes and license

Passenger Transportation

The subject of passenger transportation is receiving considerable attention now and the question is suggested for discussion: Why not use semi-trailers with passenger bodies? They would

(1) Accommodate more passengers
(2) Have a lower center of gravity, be easier to get on and off from and be safer to ride in
(3) Ride easier than four-wheel buses
(4) Be quiet to ride in
(5) Be free from engine and oil odors
(6) Have a cost per seat-mile less than one-half of that of the usual type of bus

THE DISCUSSION

Phil Overman:—Do the regulations Mr. Greer mentioned, governing power, pressure and speed refer to all types of paving in the State, to dirt roads or to concrete paving?

E. R. Greer:—Those are about the only regulations we have in Minnesota now, and they are the regulations now being advocated for a National vehicle law. I believe the Minneapolis city ordinance agrees with them at present; there is no distinction made as to whether it is a city pavement or a country road.

C. S. Moody:—In regard to ordinary hauling with trailer equipment, how is that limited?

Mr. Greer:—We are running 3½-ton trucks and 3½-ton tractors of exactly the same make; they have the same size engines, the same kinds of traction and axles, and the same sizes of gears. In one case we are carrying a 3½-ton load and in the other case we are normally carrying from 10 to 12 tons. A comparison of these units shows an average of about 4½ miles per gal. with the truck, and 4 miles per gal. with the tractor and trailer. In other words, the difference is less than ½ mile per gal. We get an extra ½ mile per gal. when we pull 3½ tons instead of 12 tons and the operating conditions are pretty severe right now. We hauled greater than 12-ton loads of brick to building jobs during our deepest snow and put them in, in a section where nobody had driven before. We would get in and get stuck in the snow; so we had to back out and pull ahead a few times, but none of our drivers got stuck for more than 20 min., even with the 12-ton load.

Mr. Moody:—Have you had more or less trouble with the 3½-ton trucks?

Mr. Greer:—The trucks have been stuck just as often as the tractors with trailers, and they have been just as long in getting into and out of the snow. The point is that the load on the wheels for the size of tires is about the same, and very little more power is required to pull the additional load on the trailer wheels. Of course, the trailer can get cramped around where the trailer wheels do not track with the tractor wheels; so, the trailer wheels might be in a snow bank and the truck wheels out of it. Under such conditions the tractor wheels would spin around idly.

The four-wheel-drive truck advocated now for rail use is a similar trailer proposition. This truck is built into a tractor, and the body that the passengers ride in is a trailer body which is carried partly on a power unit. In the original test, it required eight box-cars to constitute the limit of traction that could be maintained with a steady pull by the truck. On a level track, the dynamometer registered a 3600-lb. pull to start those box-cars into movement. After they were well started a 3000-lb. pull hauled the total load, which was 120 tons. That was on dry rails. When the rails were wet, it was impossible to pull anywhere near so great a load. That truck weighed 6300 lb. and had a 4-ton additional load; so there was somewhat more than a 10,000-lb. weight on the traction wheels.

Mr. Moody:—On this power-trailer proposition it seems to me that this can be confined only to smaller units, unless it is run on rails. By running it on rubber tires, the steering proposition would not only be complicated; it would be necessary to have differentials. These difficulties could be eliminated by running the car on rails, because the car would steer itself; it would not require any steering mechanism or any differentials to be incorporated in its equipment.

Questions Answered by the Research Department

MANY of the inquiries answered by the Research Department are of much general interest and involve a considerable amount of thought and research. For the information of our members a few of them will be published each month. It is hoped that in this way those members who have not yet availed themselves of the facilities offered by the Research Department may be encouraged to take advantage of them in the future.

COAT-FINISHING EXHAUST-MANIFOLDS

THE following inquiry was received from one of our members, a British builder of high-grade automobiles:

> We are experiencing considerable difficulty in obtaining a good, high-class, permanent finish and appearance on the exhaust-manifold of the engines of our cars, and would be pleased if you could supply us with some information on the various methods used by firms in the United States to obtain the high-class finish they do.

A questionnaire based upon this inquiry was sent out to a number of producers of high-grade cars and to a number of enameling companies to whom the car builders referred us for further information. The following extracts from letters received from the automobile companies summarize their views on the question:

> We have found two very satisfactory ways of finishing our exhaust-manifold. One is Parkerizing with a black Bauer-Barff finish of a fairly permanent nature. The other is a vitriolizing process that enamels the exhaust-pipe with a glassy compound of great heat-resisting qualities and effectively prevents rusting.

> The porcelain enamel finish represents the very best obtainable finish that will withstand any amount of heat.

> We wish to emphasize that we have had considerable difficulty with our exhaust-pipes so far as keeping a reasonable finish on them is concerned. We have tried metal coating and all sorts of enamel, but nothing is satisfactory. Lately we have tried fusel oil, which we find gives a nice finish and withstands considerable heat. We have been in the habit of sending out our work to an enameling company which subjects the exhaust-pipes to a treatment similar to that applied to pottery. The coating is first a dull luster and after being subjected to about 1200 deg. fahr. becomes a brilliant glaze. It looks very much like black enamel and maintains its luster very well up to a temperature of about 1400 deg. fahr.

> We have found after considerable investigation that fused enamel similar to that used on the best grade of cooking utensils is the best suited for the permanent coating of exhaust-manifolds. All other finishes are more or less unsatisfactory, and they do not present so good an appearance as the fused enamel.

The enameling companies replied, in part, as follows:

> There are two methods of porcelain enameling. These are known as the dry-coat and the wet-coat processes. It is by the latter that the automobile companies in this country are finishing their manifolds. They are using this process because it gives them a much more durable finish and at a considerable saving. The method employed for finishing is as follows: The casting is first sand-blasted to give it a smooth finish and to remove all dirt or foreign substances; the enamel is then sprayed on in liquid form and allowed to dry thoroughly, after which the manifolds are placed in a furnace at a temperature of about 1300 deg., which fuses the enamel in with the iron and gives the desired porcelain enamel finish.

> We have a special finish for exhaust-manifolds that consists of a high-gloss black coverage applied at a temperature of approximately 1400 deg. which is nearly double that of the exhaust-pipe of an engine in ordinary use. This gives practically a permanent finish.

> The method we follow in enameling exhaust-manifolds is as follows: The manifolds are first rough-ground so that all protruding chaplets, fins, etc., are removed. The manifold is then sand-blasted with flint shot so that it has a silvery white finish. It is essential that all burnt sand be removed from the casting. The casting is then sprayed with a special black manifold-enamel which is allowed to dry for about 3 hr., at the end of which time it is put in a gas-fired muffle furnace. The temperature in the muffle is approximately 1350 deg. fahr. It takes about 10 min. to fuse the enamel onto the iron so that when the manifold is taken from the muffle it has a high gloss. We find it necessary to respray this manifold and fire it again to get a first-class smooth job. After the manifold is sprayed the first time it is necessary to wipe some of the joints where the exhaust from the cylinders enters the manifold, so that when the manifold is put on an engine it can be fitted to the cylinders properly with a gasket.

> We finish exhaust-manifolds made of malleable iron with one or two coats of black vitreous enamel. The process of vitreous enameling that we use is called the wet process. That is to say, the article to be enameled is thoroughly cleaned, first being subjected to a high temperature. Then it is sand-blasted, after which we spray either one or two coats of enamel upon the surface of the article and burn it at a temperature of from 1400 to 1500 deg. fahr. This gives a fine finish which will withstand a temperature of 1500 deg. fahr., but if subjected to a constant temperature above that, it is apt to lose its finish, become dull and possibly blister. Enameled manifolds should not be used on automobile engines that are being tested.

> Most of the protective finishes used on exhaust-manifolds proved more or less unsatisfactory until the development of vitreous enamel finish, for cast or tubular exhaust-manifolds, in recent years. This finish appears to have given highly satisfactory results as to both appearance and permanence.

Discussion of Papers at the Detroit Production Meeting

THE discussion of the papers presented at the Production Meeting of the Society held at Detroit, Oct. 26 and 27, is printed herewith. Efforts have been made in every case to have the authors of the various papers reply to the discussion, both oral and written, and these comments, where received, are included in the discussions. For the convenience of the members a brief abstract of each paper precedes the discussion with a reference to the issue of THE JOURNAL in which the paper appeared, so that those who desire to refer to the complete text as originally printed and the illustrations that appeared in connection therewith can do so easily.

EXPERIENCE NOTES FROM A PRODUCTION NOTEBOOK

BY H. J. CRAIN AND J. BRODIE

WHILE investigating the sources and causes of noise in automobiles during an extensive connection with one of the largest automobile companies, the authors recorded their experiences in the shop in the form of notes. Some of these are offered with a view to stimulating the discussion of the subject and with the hope that additional information will be brought out by an exchange of ideas, particularly on the problem of eliminating gear-noises. In many cases they found that noise was caused by failure to allow sufficient clearance for an adequate oil-film. And it was noted frequently that when one noise had been located and silenced another appeared that had not been apparent before. The topics that have been considered include the running-in of brake-bands, engine knocks, oil-pump gear-noise and that of gears in general, the clearances of ball bearings, backlash, and rear-axle bevel-gears.—[Printed in the November, 1922, issue of THE JOURNAL.]

THE DISCUSSION

PRESIDENT B. B. BACHMAN:—I have had a suspicion that has been confirmed again and again, that we all have a fetish for close fitting, that we forget many times that oil and oil films are actual, physical things and require space and that, with bearings that just fit, when there is no clearance space between them even for atmosphere, the possibility of getting lubricants into them is reduced to zero.

H. M. CRANE:—I am very much interested in the introduction of loosely fitted gearbox bearings to produce quiet gears and especially in connection with the use of ball bearings. Has any test been made of ball bearings in which the balls and races are fitted tightly but the outer races are fitted loosely in the housing? I do not know that that would give the same result. It probably would not, but I think that it might give the desired cushioning effect and the same quieting result.

R. F. RUNGE[1]:—Referring to Mr. Crane's question, I do not believe that the same results could be obtained by making a loose fit between the outer ring of the ball bearing and the housing. This form of construction may tend to dampen some of the transmitted noise but would be dangerous, as the tendency would be for the outer ring to creep or spin in the housing. This would cause excessive wear, especially where the housing is made of aluminum.

The chart shown as Fig. 8 in the paper is apparently based upon theoretical calculation and is evidently intended to give only an approximate idea of the comparison between axial or end play and radial looseness. Actual measurement of the end play shows that for a specific amount of radial looseness the end play will be from 25 to 30 per cent greater than the amount indicated in the chart. The determination of the radial looseness by end-play measurements is not at all reliable, as the amount of total movement of one ring as compared with the other depends on the accuracy of the contour of the ball grooves, the radius of the grooves, the amount of pressure applied when the measurement is taken and whether the one ring is moved in a plane parallel to that of the other ring.

Considerable variation can be had in all of these items but it is at the same time possible to hold the radial looseness constant. In all of the variations that are encountered, the one that will have the greatest range will come from varying the pressure applied when the end play is measured. Under a very light load, there is excessive end-play due to the elastic deformation occurring at the point of contact of the balls with the grooves. As a result, we have found that where the amount of end play must be considered, its determination can be made only when a specific end-load in either direction is applied. From this explanation it will be seen that the determination of radial looseness by the amount of end play is not very reliable. We have developed machines that will determine either radial looseness or radial tightness within 0.0001 in.

Another consideration that is not always taken into account is the amount of looseness or tightness remaining in a bearing after it has been mounted. We find, from experiments, that where an inner race has a press fit on the shaft, the ball groove of that race will expand approximately 70 per cent of the amount of the press fit. Therefore, considering that a bearing has a radial looseness of 0.0005 in. and the inner race has a press fit of 0.0010 in., after the bearing is mounted it will be actually tight by 0.0002 in., all of the looseness having been taken out by the press fit and 0.0002 in. of tightness put in. Where it has been predetermined that a certain amount of radial looseness is desirable, it is very important to be sure that the amount of final looseness required has not been interfered with by the mounting of the bearing. As it is not general practice to force the outer ring of the bearing into the housing, it is not so important to consider these same conditions as applying to the outer ring, although in such cases as where the outer ring is

[1] M.S.A.E.—Vice-President, S K F Industries, New York City.

forced into the housing, that should be taken into consideration. It should also be kept in mind that where those types of bearing that must be opposed to stabilize the position of the shaft are used, it is practically impossible to maintain radial looseness or to know its magnitude.

A tightly fitted bearing will make more and a different kind of noise than one that is loosely fitted. The tight bearing will also set up a greater noise where inaccuracies in the bearing exist or where foreign matter, such as metallic chips or abrasive matter, is present. The noise in bearings many times is traceable to the presence of foreign matter, and extreme care is taken in the manufacture of the bearings to eliminate this condition and every precaution should be taken by the user to keep the bearings clean in handling or assembling.

Where normal or tightly fitted bearings are used, the gear noise, as well as bearing noises, is very efficiently transmitted to the gearcase or the housing which, in most cases, will magnify the noise. It is a well-known fact that where an oil cushion can be provided, it will dampen out part of the noise and vibration. I believe, however, that in the use of the looser bearing for application to transmissions, as the looseness of the bearing will more freely accommodate misalignment of shafts and their deflection under load, greater flexibility plays a bigger part in noise elimination.

G. E. GODDARD:—In his experiments with this radial play, did Mr. Brodie discover any appreciable difference in the oil-pumping characteristics of the ball bearing? In some of our earlier experiences we noticed that some ball bearings pumped oil in a certain direction. I think the problem was never solved in our own minds. We turned the bearings end-for-end and they pumped in the same direction. We tried bearings of different makes, with different ball separators, and that seemed to make very little difference. Would increasing the rolling clearance affect the oil-pumping of the bearing?

I think it is common practice to use an oil spinner or throw-off ring to prevent the oil going through a bearing, but believe it would be rather interesting to hear from some of the ball-bearing men as to whether they have had any experience with curing oil-pumping in annular ball-bearings.

If there is an existing standard on the radius of the groove in the races, is it 7/10 of the ball diameter or do the different makes of bearing have different radius grooves? If they do, would it not be necessary to have a table for each make of bearing?

R. S. DRUMMOND[2]:—We have had some experience in our factory along the same line that Mr. Brodie mentions, concerning the radial clearance of ball bearings. The work that Mr. Brodie mentions in his paper is confirmed by experience in testing gears in our factory. The tests were suggested by gears that were quiet with plain bearings and noisy with other bearings, and were made using Hess-Bright bearings of various radial clearance. I speak of this with reference to automobile transmissions only, as there appear to be other fields of usefulness for ball and roller bearings which are not as yet affected by close fit of balls.

We have upward of 15 different transmissions, representing a number of cars that are produced by men present in this room. These were furnished us by the car builders for test purposes. In every case the bearings in the transmissions require some alteration in the amount of radial clearance for them to operate quietly. Comment has been made about the amount of radial clearance or oil room needed. There are no statistics that are absolute on that subject today. The amount mentioned in Mr. Brodie's paper, 0.0008 in., is approximately correct under average conditions in average transmissions as tested by us. Where there are no means of checking in a laboratory the amount of clearance in the bearing, we have used a very simple expedient. We hold rigidly with one hand the inner race, and the cage carrying the balls and try to slip the outer race over the balls. A crude but very satisfactory estimate of the amount of radial clearance can be made in this way. If the outer race slips in that way, there is normally enough radial clearance. In four noisy transmission jobs investigated in the factories of the manufacturers, we established to their satisfaction that the radial clearance was lacking to an extent that developed noise in the bearings and that noise was eliminated by increasing radial clearance. Our tests confirm what Mr. Brodie has said.

Our experience indicates the same general result from the standpoint of backlash in gears. In cold weather the teeth operate somewhat like individual dashpots. The oil does not always get away with facility at the bottom of the tooth; it has to be squeezed out under pressure and, if there is not sufficient room for the oil to escape readily, a noise will develop. Backlash in various types of gear and in various transmissions depends to a certain extent upon the construction. There is one transmission that requires approximately 0.004-in. backlash to operate with any reasonable degree of quietness. We believe this is due to poor support of the driving pinion. When this support is improved the gears run quietly with greater backlash. Average transmissions show a requirement somewhere between 0.010 and 0.016-in. backlash.

T. C. DELAVAL-CROW[3]:—Mention is made in the paper of the effect of radial clearance in ball and roller bearings on transmission noise. The author's experience indicated that an increase of radial clearance within certain limits reduced the noise. The reason advanced for this is that sufficient space was left for an oil-film between the balls and the races. The experience of the New Departure engineers corroborates the results obtained by Messrs. Crain and Brodie but we cannot agree with the reasons advanced for the reduction in noise, as the very heavy unit-pressures at the contact points of the balls with their races absolutely preclude the presence of an oil-film in the commonly accepted sense of the term and, therefore, it seems that some other reason should be assigned for the reduction of noise.

A study of the noise problems indicates that there are two main causes, namely, noise due to faulty tooth-contact arising either from an imperfect tooth-form or from a misalignment of the transmission parts in assembly, and the secondary cause that might be traced to the inherent structure of the bearings themselves and their mounting. The trained ear can easily distinguish between the two causes of noise, that caused by the bearing being a distinct whistle and that from the gears a growl. In many cases the modification of the bearing structure and mounting will alleviate the noise due to faulty tooth-contact of form. Our experience indicates that there is no general rule as to the degree of modification of the bearing structure that will lessen noise in all cases. Each particular transmission is a study in itself, the design of the transmission housing having a very great influence on the degree of amplification of noise vibrations.

As a general thing, attempts have been made to use ball bearings with no radial clearance, little regard being paid to the effect of the expansion of the inner race due

[2] M.S.A.E.—Vice-President, Gear Grinding Machine Co., Detroit.
[3] M.S.A.E.—Chief engineer, New Departure Mfg. Co., Bristol, Conn.

to excessively heavy press-fits on shafts and to too tight fits of outer races in housings. When assembled under these conditions, the bearings themselves will have a distinctive whistling noise due to the vibration set up through the metallic parts under tension, as in the case of a piano string. Noise such as this can be decreased or practically eliminated entirely through the lessening of the press fit on the shaft and the reduction or elimination of the tightness of the fit of the outer race in the bearing housing. It is essential that the outer race be not a tight fit in its housing, due to the fact that it is practically impossible, as a production proposition, to produce housings accurately round, and the extension of out-of-roundness of the housing is transmitted to the ball race of the bearing on account of the deformation of the outer race.

An example of the influence of the housing out-of-roundness on the race ring may be cited from an experiment conducted with a light aluminum housing of an average thickness of 0.1870 in. about the bearing seat which when accurately measured was found to be 0.0006 in. out-of-round in the bore. Into this was inserted with a very light press-fit, a plain bearing ring of high-carbon chrome steel, having a width of 0.6693 in., a section of 0.2960 sq. in. and a base diameter of 2.4410 in., which was lapped to a perfect ring on both the outer diameter and the bore, and accurately measured before insertion into the housing. Upon being inserted into the housing which, as has been previously said, was approximately 0.0006 in. out-of-round, the bore of the ring was found to be 0.0004 in. out-of-round. Upon the reduction of the fit in the housing from a press to a push fit, the out-of-roundness of the ring was eliminated. The expansion of inner-race rings of bearings due to press fits on shafts is likewise often neglected. Investigations have shown that the increase in the diameter of the inner raceway is dependent on two things: the amount of fit used and the relation of the race thickness to the shaft diameter.

A study of the chart at the left of Fig. 1 will show that a press fit greater than 0.0005 in. is detrimental, because the race expansion increases very rapidly above this point, and is also liable to make trouble in assembly due to scoring of the shaft, causing insecure seating and possible misalignment. It is also seen that for a given increase in the shaft diameter the expansion of the cone raceway is greater in the case of the No. 1310 bearing than in that of the No. 1307.

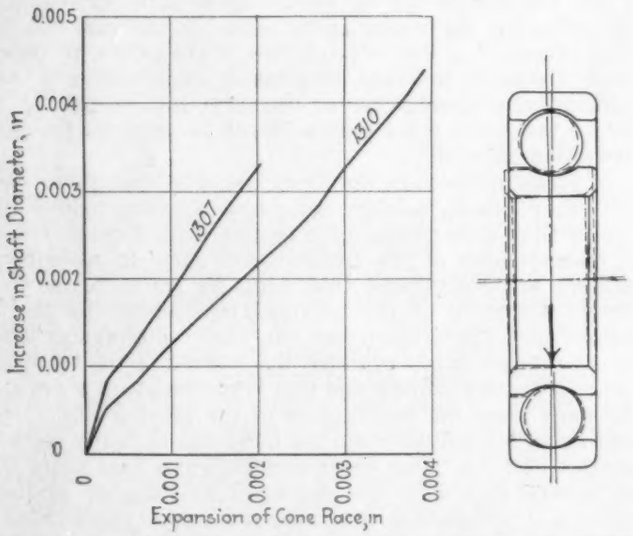

FIG. 1—THE CHART AT THE LEFT SHOWS THE RELATION BETWEEN THE SHAFT DIAMETER AND THE EXPANSION OF A CONE RACEWAY IN A BALL BEARING, WHILE THE DRAWING AT THE RIGHT ILLUSTRATES HOW MUCH IT IS POSSIBLE FOR THE SHAFT ON WHICH THE INNER RACEWAY IS MOUNTED TO DEFLECT BEFORE THE UPPERMOST BALLS MAKE FORCED CONTACT WITH THE UPPER PORTION OF THE OUTER RACE

Considering the fact that the bearing is used merely to support the rotating members and not as a means of transmitting torque, the use of an excessive press-fit is not necessary. The press fit need be only great enough to assure that the circumference of the shaft is larger than that of the cone bore, in which case no slippage can occur and consequently no wear or peening of the shaft take place.

It is the practice of the New Departure Mfg. Co. in the production of its ball bearings to allow sufficient radial clearance between the balls and their races to compensate for inner-race expansion due to average press-fits not in excess of 0.0005 in. and in addition to allow a very slight excess of clearance to take care of the possible misalignment of bearings in housings or of the transmission shafts in assembly.

A fact that is not as generally recognized as it should be is that in properly constructed ball-bearings provided with slight amounts of radial clearance there is very marked ability to operate satisfactorily under conditions of shaft and housing misalignment. This is due to the fact that in the presence of clearance with other than gravitational load, the inner race and the lower balls will establish perfect contact with the outer race at the lower side, leaving all of the clearance on the upper side between the uppermost balls and the outer race, so that, as illustrated at the right of Fig. 1, the shaft upon which the inner race is mounted can deflect about a suitable angle before the uppermost balls come into force contact with the upper portion of the outer race. Under actual running conditions the clearance between the uppermost balls is greater than under static conditions, due to the deformation of the races and the balls under load, so that the angularity of shaft deflection transmittable under these conditions is in direct proportion to the amount of load on the bearings.

It will be seen that, if the radius of curvature of the outer race approximates the radius of the ball too closely, the ability of the bearing to withstand ordinary deflection misalignments is greatly reduced, in spite of the fact that there may be built into the bearing a relatively large amount of radial clearance.

Our experience has indicated that it is possible to lessen seriously the availability of a given bearing, through the establishment of too close race curvatures, especially in the outer race, to carry ordinary deflections or misalignments without excessive noise. While, in a properly designed and manufactured ball-bearing, a limited amount of radial play is desirable for the reasons given above, it can be seen readily that excessive radial clearance may become detrimental to the load-carrying ability of the bearing in that when there is excessive radial clearance the diameter of the ball race in the outer race is materially greater than the sum of the diameters of the ball race of the inner race plus two ball diameters, causing the radial load of the bearing to be carried on two or three balls rather than on half of the balls in the bearing.

As stated by Messrs. Crain and Brodie, it is practically impossible to measure radial play. We therefore advocate that the measurement of end play under a known load be used as an indication of the radial clearance existing in the bearing. It is well to remember that this means of measuring is only an indication, as the mathe-

matical relation of the radial clearance to the end play is dependent on the percentage of the ball diameter to which the race radius is ground and a very slight change in this percentage makes a relatively large change in the end play obtained for a given amount of radial clearance. The race radius on all high quality ball-bearings is ground by a generating grinder and any change in the race diameter from the standard set-up causes an equal change in the race radius, therefore causing a variation in tne end-play relation to the radial clearance.

SOME UNIQUE FEATURES OF AUTOMOBILE PRODUCTION

BY WILLIAM DUNK

THE author discusses the Franklin automobile designs in regard to the production of an air-cooled cylinder, the making of case-hardened crankshafts and the machining of duralumin connecting-rods, commenting also upon experiments made with a view to producing hot-swaged rear-axle drive-shafts.

The unit-cylinder construction of the Franklin engine, with the cylinder-head cast integrally with the body of the casting and steel cooling-fins cast in the outer surface of the cylinder-wall, presents a different problem from that of machining cylinders cast-in-block, and the procedure is described in some detail, inclusive of the machine tools used. The case-hardened crankshaft and the duralumin connecting-rod are treated in a similar manner. The experiments in the manufacture of a hot-swaged rear-axle drive-shaft have extended over the past 18 months, but have not yet justified quantity production by this method.—[Printed in the November, 1922, issue of THE JOURNAL.]

THE DISCUSSION

K. L. HERRMANN:—What percentage of cylinder scrap does the Franklin company have in connection with its production of cylinders?

W. H. WORTHINGTON:—In connection with bearings, what lubricating system is employed? Of what material are the bearings made? Is the bearing surface of bronze or of babbitt metal? What clearance for oil is provided in the selective reaming?

J. H. CLARK:—Are oil-grooves used in connecting-rods and camshaft bearings?

R. H. WEINERT:—Why is it necessary to produce a smooth finish on the axle-shafts? Is it because this finish is necessary for some of the operations? What effect does scaling have? On the axles that we produce we always have scaling marks, but we find that they do not affect our production in any way.

E. PLANCHÉ:—What percentage of rejection does the Franklin company have on duralumin connecting-rods? Does it have any breakage?

ROBERT K. JACK:—Is any trouble experienced in cold weather due to this type of engine cooling off very rapidly when it is stopped, on account of the absence of water around the bores, which, on a water-cooled engine, retains its heat for a considerable length of time. Is it necessary to carry a larger battery than would be required for a water-cooled engine of the same displacement, due to the extra drain on the battery from repeatedly cranking a cold engine?

T. J. LITLE, JR.:—What is the compression pressure of the Franklin engine? How much weight was saved by the use of the duralumin connecting-rod instead of the former steel rod?

PRESIDENT BACHMAN:—I am interested in the portions of the paper relating to the case-hardened crankshaft, scleroscope values of 80 to 100 being mentioned. Has any of the experimental work indicated a remarkable difference between these values and values in the neighborhood of 70? What has been the experience with regard to copper-plating surfaces of the crankshaft? Are any preliminary finishing operations necessary to get a reasonably smooth surface on the crankshaft for that purpose, or does the ordinary finish cover the matter satisfactorily?

B. M. LEECE:—I request information particularly in regard to the hardening die used on the crankshaft.

MR. HERRMANN:—In the Studebaker plants we roll axle-shafts instead of swaging them. We use no die in rolling them, we require but one heat and the machinery stands up and gives us no trouble at all. The swaging of the crankshaft may be a different proposition entirely.

H. P. HARRISON[1]:—In answer to Mr. Herrmann's remarks, I would say that our engineers have designed a rear-axle drive-shaft that is extremely light. For this reason they insist on a perfect circular section of the shaft and a perfect taper from the large to the small end. We have not been able to convince them that we could produce a rolled shaft which would be good enough to meet their requirements.

Foundry scrap, in connection with the cylinder, averages about 10 per cent. We realize that this is rather a small figure, but it is made so by the fact that the cooling fins are inserted in the cylinder wall only 1/16 in. Should we insert them any deeper than that, we would surely have trouble with cylinders splitting in service on account of the extremely thin wall left between the bottom of the flange and the bore of the cylinder. The amount of our machine-shop scrap runs very low; 1 per cent will cover it at all times. I believe that this is due primarily to an excessively good inspection of the cylinder castings and the comparatively simple machining operation that the unit cylinder requires.

We use a pressure system of lubrication with a central oil-pump, carrying a pressure of about 80 lb. per sq. in for the main and crankpin bearings. The bearing metal is a hard type of die-cast babbitt. We do not use what are considered true oil-grooves in the Franklin bearing, but simply a small pocket that is cast entirely through the back of the bearing, making an opening about 5/16 in. wide and ½ in. long. This connects directly with an oil lead from the pump.

The oil clearance that we use with the selective method of reaming is 0.001 in. We find that this gives a very quiet engine and at the same time a new engine is free enough to start easily.

My answer to Mr. Herrmann explains somewhat the reason our engineers insist on a smooth finish on the rear-axle drive-shaft. We are working so close to the ultimate strength of this shaft that we cannot have the least suspicion of an abrasion in the surface of the shaft at the neck. The scaling that I mentioned seems to have the

[1] Production department, H. H. Franklin Mfg. Co., Syracuse, N. Y.

effect of making rather deep abrasions during the swaging process.

We have never had a duralumin connecting-rod fail in service. This is rather a remarkable record for putting a new part like that into an engine but, as yet, we have experienced no trouble whatever from this source.

In regard to starting the Franklin engine in cold weather, due to the excessive cooling effect of the fan as soon as the engine is started up with the starting motor, we use an electric primer in connection with the carbureter; it is used to warm the gasoline itself before it passes into the venturi tube and so on into the manifold. This has the result of vaporizing the starting charge quickly. We experience no trouble whatever in starting in cold weather.

The pressure cooling-system we are using now allows us to carry a compression pressure of about 4 atmospheres, which gives us considerable increase in power over that of the engine we have just abandoned.

The relative weights of the steel connecting-rod formerly used and the duralumin connecting-rod we are using now are as follows: the steel rod weighed slightly more than 2 lb.; the duralumin rod weighs but 13 oz. This decreased weight has the direct effect of reducing the amount of engine vibration. The steel rod was designed for a crankpin bearing of only 1⅝-in. diameter, while the duralumin rod was made to take a 2-in. crankpin bearing.

We have had some shafts go through which scleroscoped as low as 75 at some points, and they gave us no trouble at all. The principal reason for keeping the requirements as high as they are is to endeavor to reach the maximum rather than to lower the requirements, which might induce the heat-treating department to make an insufficient effort to get a shaft that can really be called hard.

In preparing the crankshaft for copper-plating, we do not find it necessary to machine the surfaces that are to be plated. The forging is simply blasted with metal-abrasive shot, and the copper-plating is applied directly to the surface. We have had no trouble due to failure of the copper to adhere properly to the forged surfaces. It is necessary to have the copper adhere only long enough to take the shaft through the carburizing process.

The hardening *die* we use is an evolution of many ideas that we tried out. It operates exactly like a pair of hinged doors that have V-blocks on their inner surfaces. The V-blocks are made of high-speed steel and have contact points on the bearing surfaces only 3/16 in. wide. There are eight of these contact points on each bearing, and they cover a very small area when compared with the total bearing area of the bearing in question.

SOME CAUSES OF GEAR-TOOTH ERRORS AND THEIR DETECTION

BY K. L. HERRMANN

THE different gear noises are classified under the names of knock, rattle, growl, hum and sing, and these are discussed at some length, examples of defects that cause noise being given and a device for checking tooth spacing being illustrated and described. An instrument for analyzing tooth-forms that produce these different noises is explained and illustrated.

Causes of the errors in gears may be in the hardening process, in the cutting machines or in the cutters. A hobbing machine is used as an example and its possibilities for error are commented upon. Tooth-forms are illustrated and treated briefly, and the hardening of gears and the grinding of gear-tooth forms are given similar attention.—[Printed in the November, 1922, issue of THE JOURNAL.]

THE DISCUSSION

J. F. RAND:—Were the gears that we have been discussing finished in one cut? If not, how much material was left on them for the finisher? What type of machine finished them, a hobbing or a gear-shaping machine? We find it advisable to attempt to build a special machine of our own to overcome the indexing trouble mentioned.

O. LINDBERG:—We have found it impossible to produce an accurate gear with a gear-hobbing machine. We have used gear-hobbing machines only for roughing purposes, and use a gear-shaping machine for finishing.

C. F. SCOTT:—Many of the British and French cars are built with relatively small engines for economical reasons. The cars are fitted with four-speed gearboxes and much gearshifting is necessary, which might not be acceptable in this Country. These gearsets are, some of them, direct-drive on third, and some drive direct on fourth speed but, in any event, at one of the two top speeds they would be comparatively noisy. Would the noise at either top speed be considered excessive in accordance with American standards, or is the transmission fairly quiet? If so, how do they make it quiet?

MR. RAND:—What is Mr. Herrmann's opinion regarding just how far one can go with spiral angles? In other words, suppose a 30-deg. spiral angle on a given diameter of gear of a certain pitch, is it an advantage to go over 30 or 35 deg.? What is the limiting factor on that spiral angle?

A. J. BAKER*:—The essential point Mr. Herrmann brought out is how difficult it is to maintain the proper operation of the gear-cutting machine. When it is considered that this machine, no matter of what type, is composed of two rotating members, each subject to eccentricities due to the initial errors brought about during their manufacture, and that those errors can be added to by temperature conditions, stresses, material variations and relative dullness of the cutting portions, we are forced to the conclusion that the whole principle is wrong.

It is of little use to make a statement of that kind unless some constructive criticism is offered. Do the gear-cutting-machine builders give any consideration to the use of a single-point finishing-tool, such as a broach, in conjunction with the rotating work, so as to eliminate all of those points that give trouble? I have in mind that a single-point tooth for the actual finishing tool can be made far more accurately than it is possible to produce a number of cutting tools, each of which has a different relation to the true axis of rotation. That is applicable, I believe, to only a straight spur-gear, but we would all be happy if we could eliminate some of our spur-gear troubles.

MR. DRUMMOND:—The indicating apparatus shown by Mr. Herrmann is well known to me. I admire the work

* Research engineer, Willys-Overland Co., Toledo.

that he has done. He is more or less a pioneer in the study of tooth-form. The efforts to develop mechanisms for measuring gear teeth are very worthy. Many types of measuring apparatus are being offered today. In the interest of good gearing, they should be given careful attention because they are worthwhile.

I am also interested in grinding operations and in tooth-form error or distortion. The transmission Mr. Herrmann is working on is one of the most difficult transmissions in America to make quiet. It is a combination of 16 and 32-tooth gears and a rear gear having 13 teeth that drives a 14-tooth gear. The teeth are stub teeth and the pressure-angle is 14½ deg. If you will figure it out and then cut some of these gears in your own factories, I think you will look at the charts again with more interest and pay greater attention to the subject of measuring gear-tooth form. The transmission in question is hard to build, but this is being done successfully.

Mention was made of a machine that would have a broaching action. Gear teeth can be and have been broached. We are using the shaping-type machine with a grinding-wheel form to fit the two sides, which is more-or-less similar to the broach in its action. Regardless of our own individual method of doing this work, we think that the future of the industry lies in the direction of doing final work after the distortion takes place. We are not opposed in any way to doing it by any other method, but I think there will be difficulty in making hardened gears until we do final work after distortion.

The variation in gears that Mr. Herrmann speaks of sometimes takes peculiar form. We have on record a gear that had a number of teeth, going half-way around the gear, with an index error as high as 0.015-in., but gear noise in the transmission was remarkably slight.

MR. HERRMANN:—It is true that grinding can be resorted to. We have had rather wonderful success with such a machine for the past 2 months. In this particular grinding process, however, it is necessary that the gear be fairly accurate; otherwise the output of the machine is very limited. We are getting gears with less than 0.4-per cent rejection when ground on this machine. However, grinding alone does not solve the problem. We do not need to harden some gears. Some gears are spiral. They involve a somewhat more expensive process than we can afford to use in connection with a medium-price car. Spiral-bevel-gear grinding is unknown to us at this time.

The gears mentioned in the paper were roughed-out first and then finished in a more accurate machine that is more carefully maintained. It is possible to hob an accurate gear. Gears can be hobbed as accurately as they can be cut on a shaping machine.

The practice in Europe is entirely different from that in America. Some years ago I worked as a toolmaker at the Napier plant in London. When producing only a few cars per week, many things can be done that cannot be done in large production. European gears are much quieter. The work put upon them makes them so.

I do not know whether a 30-deg. spiral angle is better than 45 deg., which we use to a large extent. The spiral angle, as well as the pressure-angle, and tooth heights of full or short length, seem to have very little to do with the matter if the other errors are within proper limits. I prefer the steeper angle.

As to the necessity of a new gear-cutting machine, most of the important gear-cutting-machine builders are designing new machinery.

Silent chains help to reduce noise in a motor car. They offer many problems. I refer you to the service divisions of the companies that use them. Accuracy of gear-tooth form is necessary; otherwise silent chains are noisy. The same principles that have been discussed here in connection with gears can be discussed in connection with silent chains.

After we do the best we can with gears and get them just as nearly right as we can measure, the gear situation costs the motor-car manufacturers $11,000 per day. We estimate that this figure represents the lowest possible amount that is being spent on this problem.

A. B. REYNDERS:—One of the principal reasons for using plain bearings on lighting generators is the quieter operation in comparison with ball bearings running at high speeds.

Owing to the fact that no absolute standard of noise is available, there are about as many different opinions with regard to quiet operation as there are persons judging the apparatus. To eliminate this difference of opinion we should have some sort of a noise-measuring device or some kind of a standard. At present we select a machine which, in the opinion of a number of people, is considered quiet. This machine is set aside as a standard and production is obtained by making comparisons with this standard. Is a more accurate method available at the present time in the manufacture of automotive parts?

MR. PLANCHÉ:—It is a recognized fact that gear noise is very intimately connected with the pitch sizes of gears. A noise is created by a metallic vibration. This metallic vibration is produced by the impact of one gear tooth against another. The greater the pitch for a given diameter, the greater is the interval between each impact for the same speed. I am sure that, if the engineers would turn their thoughts to the utilization of a smaller tooth-pitch, widening the gear if necessary to preserve the strength, many of our gear problems would be eliminated or greatly decreased. To confirm this opinion, we have only to study the steam turbine in which the use of small-pitch spiral-type gears of great width has been very successful in transmitting power with the minimum amount of noise.

THE GROUP-BONUS WAGE-INCENTIVE PLAN

BY E. KARL WENNERLUND

THE author states that the purposes of every plan of wage incentive are to stimulate the worker to a greater effort than is generally obtained on a straight day-work basis; to reward him somewhat in proportion to his effort; and the gaining of other advantages such as greater attention to conditions that curtail production, more uniform labor costs and the elimination of inefficient employes. He states further that nearly all industries engaged in repetitive work are now on an incentive basis.

After outlining the most successful wage-incentive plans and enumerating some of the conditions that must be met, inclusive of four specific fundamental principles of industry that are stated, the group-bonus plan is explained and the application of group standard-time is discussed at some length, supplemented by tabular data. Experience with grouping is then related and conditions favorable to grouping are mentioned. The summary states that the group plan, like any other wage-incentive plan, has for its main objec-

tive intensive production.—[Printed in the November, 1922, issue of THE JOURNAL.]

THE DISCUSSION

CHAIRMAN H. W. ALDEN:—What effect has this group-bonus wage-incentive plan had on the inspection problem in the plant? Has it decreased the ratio of inspectors to producers and improved the general quality and output of the product?

A. J. SCAIFE:—How is the individual rewarded in each group? How does the reward of the man old in the service compare with that of the man just employed, and how do the rewards of the efficient and the inefficient workers differ?

JAMES H. MARKS:—What procedure is followed with respect to his bonus in the case of a man who is discharged or quits during the pay period?

We also have installed a group-bonus system in our plant that is very similar to that of Mr. Wennerlund, although I believe that our groups as a whole are somewhat larger than his, as we have grouped entire departments. The basis of bonus that we use is certain standards of efficiency that we have determined for machine operators, assembly men, molders, steam-hammer men and others. We have set various standards of efficiency rather than use one of 80 per cent as described by Mr. Wennerlund. We offer a 1-per cent bonus for each point that the group efficiency exceeds the predetermined standard of efficiency. We take care to have varying rates per hour for the different employes in the group, in proportion to their speed and ability. What is meant when the efficiency of the group exceeds 100 per cent?

S. W. TAYLOR:—I worked for Mr. Wennerlund for a number of years. We based our time-studies on 100-per cent production for the good average worker.

J. C. WILLIAM SMITH:—At the Willys-Overland plant in Toledo we use a similar plan, but it is straight piece-work and each man in the group is assigned a specific piece-price that is paid him for each operation he performs. For example, in our gear-housing group there are eight Gisholt machines, together with numerous other machines; if the group turns out 500 housings at the end of the line, for the day, we pay each man in the group for 500 housings. Where there is more than one machine on the same operation, as in the case of the eight Gisholts, the pay is divided equally between the eight men, if agreeable to them. If not, each man is paid for the number of housings on which he performed the operation, but the total paid for does not exceed 500. The foreman makes the decision as to the number, if the men do not agree. In this way we have done away entirely with the job tickets mentioned by Mr. Wennerlund. The method obviates paying for the gear housings before they are actually received. It gives us the individual efficiency of each operator and reduces the number of our inspectors, checkers and truckers, as the work does not touch the floor. It demonstrates to each man that he is paid exactly for the work performed. We use guaranteed piece-work prices and the price is not changed before the end of a year.

W. G. CAREINS:—I am connected with the Milwaukee plant of the Nash Motors Co. We use group-bonus plans. We use some straight piece-work and, in fact, we are not partial to any one system, but use what we think is best adapted to the particular line of work in hand. In case a new man comes into a group to perform an operation that requires some time for him to become proficient in it and to develop a standard rate of production, such as the instance cited of the Gisholt machines, we would pay him the day rate for a certain length of time, perhaps 1 week, 10 days or 3 days, depending upon the length of time needed for him to become proficient. The other men in the group take the balance of the pay and split it among themselves.

In some departments where we use the group-bonus plan to provide an incentive to different classes of labor, we have what we call Class A, Class B and Class C men who take different proportions of the total bonus-price. We find that this works very satisfactorily; for instance, in engine tests, where a certain number of men probably handle the hoists, another class of men who are real engine mechanics do the class of work they are fitted for and a third group run the engines. In our Kenosha, Wis., plant, every operation is individual piece-work. This plant has never gone into the bonus plan or the group plan in any way.

H. K. REINOEHL:—Must the group be kept small so that the individual members can watch each other for the purpose of eliminating the poor producer?

E. BOUTON:—In a group-bonus plan, how are costs maintained level? There is a fluctuation of cost. When does Mr. Wennerlund arrive at a true cost, or an actual cost?

J. L. LANNEN:—What disposal is made of scrap?

J. A. MURRAY:—We have in the General Motors Truck Co. plant the group-bonus system, which was installed under Mr. Wennerlund's supervision. We have been operating this system for about 2½ years and I cannot say enough in its praise. It relieves me of a great amount of detail that we had to work out when operating under the individual-effort system and has reduced considerably the amount of clerical work in both our time-keeping and production departments.

We have approximately 525 machines at work on the group-bonus plan and they are split into 23 groups. Each group is really a plant in itself, inasmuch as there is a machine in each group or division for each operation and the work flows down the line progressively, passing only from machine to machine and never jumping across from one line to another.

We also maintain what we call a day-work well. This is built up when we install a line of machines or route new parts over the line. It naturally follows that during the trying-out of the group and the demonstration of tools, a number of parts are made which we pay for on the day-work basis. This number of hours is always held back from the operators and constitutes what we call a day-work well. We consider this well the same as a machine, the company owning it, inasmuch as we already have paid for the work done on the day-work basis. Our aim is to pay a bonus only for the parts actually produced on bonus. This well, as you will see, needs to be built up only once, and we maintain it until a group is either done away with or a new well is established.

E. KARL WENNERLUND:—The effect of the group-bonus plan upon inspection methods is that inspection must become more exacting as soon as an incentive plan is inaugurated; however, that is true of every plan. For example, the highest type of car might be built on straight day-work and some inspection would be necessary; but if that same car were built under a wage-incentive plan it would be done to speed-up production. Just as soon as production is speeded-up, the tendency, at least, is for the quality of the product to deteriorate. I do not wish, however, to convey the impression that the group-bonus system alone would cause this tendency but, with the factory organization remaining the same, additional inspectors are needed to counteract it.

In regard to inspection under the group-bonus system as compared with the inspection of individual piece-work, we have gradually developed certain practices. Under either plan there is the same inherent tendency to speed-up production and for quality to go down; so every piece or a certain percentage of pieces must be inspected in any event. Inspection is not eliminated or quality improved simply by introducing groups, but we are coming to the point where we pay for the good product only, unless it can be shown that the material from the manufacturer was defective or had defects when it came to the department. The result is that the group tries to turn out better work instead of trying only to get the work through.

For example, in a plant where we are introducing the group-bonus plan now, armatures for starting systems are tested and a certain percentage of them fails to pass inspection. It is very difficult to say that this is the fault of any one employe. Formerly, the armatures were laid aside and, when a certain number of them had accumulated, they would be sent through on a day-work ticket to be fixed. It was not practicable to deduct the cost from anybody's pay, because it was impossible to determine who caused the trouble. We told these employes that we would pay only for the good armatures and that we had found the defective ones to constitute about 2 per cent of the total number. Therefore, we said we would add 2 per cent to the time standard, so that, if the defects ran higher than that, there would be a fine against the men. On the other hand, if this proved to be unfair, we would adjust it. Now, when a defective armature comes through, the tendency is for the inspector to throw it back into the line, and we do not have to get the elapsed time for fixing it. We used to get time tickets that were charged to expense.

Defective products fall into one of two classes; the first is defective work that can be corrected, and the other is simply junk. If the product has to be scrapped, this is done just the same as under the individual-effort system. If it can be fixed on the line, it is sent back without any ticket and without any credits. That plan has had a very good effect in standardizing our defective work.

As to the division of earnings, there may be a man in the group on an operation that requires great skill, the result of years of service, and he may have to work to limits that are within 0.0005 in. In the same group there may be a boy who drills little pin-holes on a vertical drilling machine. He places the piece in a jig and could not spoil it if he tried. Naturally, the skilful man should not have the same earnings as the boy. In this particular case we paid the skilled man a base rate of 60 cents per hr. and the boy 30 cents per hr. The efficiency of that group at the end of the pay period might be 110-per cent production. If so, each member of the group would get a 32-per cent bonus on his earnings. If each worked 100 hr., the mechanic would get $60 flat rate with 32 per cent added and the boy would get $30 flat rate and 32 per cent additional. That plan takes care of all of our division of earnings. Sometimes we have girls working on a small bench-operation and we may have a skilled mechanic on the very next operation; it would not be fair to them or to any one to divide their earnings equally.

The case of the worker who is discharged, or who quits, was a little troublesome at first. We used to tell them we would give them straight pay and that at the end of the pay period we would give them the bonus they had earned. Sometimes such men left and did not know where they were going, so that plan did not work out very satisfactorily. Furthermore, our State law is rather strict; it requires that, if a man is discharged, he must be paid that same day. We adopted a very good working rule by taking the average efficiency of the group up to the end of the previous day, and figuring the bonus on that basis for the workman who leaves the service. For example, suppose the pay period to have extended 5 days and that the workman quits today at noon. He might be discharged; if so, we are compelled to pay him his actual flat-rate multiplied by the hours he has worked, the same as for straight day-work. But we are under a moral obligation, at least, to pay him a bonus; so we take the efficiency up to the previous night. It may be 90 or 100 per cent, and it may not be the same as at the end of the pay period but, so far as the company is concerned, it follows the law of averages, being sometimes a little over and sometimes a little under. It seems to be entirely satisfactory to our men, and we have had no complaints. A man seems to think it is entirely fair if he gets his straight pay plus the bonus up to the last night on the basis of average efficiency; so it has worked out very simply.

Concerning the size of a group, we must all concede that the individual-effort incentive is right, and we find that we must secure the individual interest of the employe in his own effort. He must be placed right next to his team-mates and on similar work. In isolated cases one can do almost anything, but certain fundamental principles apply to groups and one cannot get very far away from them.

We started a complicated job recently at one of our large factories that employs about 2000 people. The first operation was performed by about eight girls each of whom did exactly the same thing, picking up little bars and measuring them with micrometer calipers. The next major operation was assembling, done by probably 8 to 10 persons, who put a bushing in and fastened it. The next group, 8 to 10 men, did the machine work. It would be a mistake to put all of these operatives in one gang or one group. The girls on the first operation knew nothing about the machine work. Assuming that the group became reduced from eight to seven because of one who quit, seven operatives would get the pay of eight. If this amount were spread out among a lot of other operatives, the group would not get the benefit of it; so, we try to keep our groups fairly small, to localize the interest. When we wanted to group the girls who performed the first operation, they objected on the ground that two of them were slow operators; so, the slow ones were replaced by faster workers.

The primary object of the group-bonus plan was to reduce the amount of clerical work. We found later that this is only a minor consideration. The big effect is in increasing production because the slow workers are eliminated, and those who remain help each other out on certain operations, all along the line.

As to what is meant when the group efficiency exceeds 100 per cent, the question should be: "What do you mean by 100 per cent?" It means whatever meaning is given it. The standard time for an operation is the time taken repeatedly by the average competent workman, over an extended period and eliminating lost and waste time. It is not the time in which the operation can be done by an expert. The efficiency measurement is the standard time divided by the actual time, and 100 per cent on production is the average time made by the average competent workman. For example, a good workman ought to dig a certain number of feet of ditch in a day; another man

may dig only half as much and a third man may dig 25 per cent more; the efficiency measurement is from an assumed standard that we called "standard time."

In regard to new men who enter the group, we found it a good plan to carry a new man for 3 days and at the end of that period to throw him automatically into the group unless the superintendent asks specifically to have the probation time extended. The effect is that the company carries the new man for 3 days and the men in the group carry him for the following 3 days; after that it is up to the man to make-good in the group.

Another method which, in principle, amounts to the same thing, is to charge half of the new man's time to the group and the other half to the expense account and not to allow the man to participate in the bonus while he is on that basis. The effect is that the group carries only half of the new man's time, and he does not get the bonus and therefore tries to enter the group. That works out very well.

The question regarding the proper size of a group and why it should be small has to do with individual effort. The worker should be able to see that the output depends upon his own efforts. As an illustration, we had 20 men snagging castings. Handling this work on an individual basis was very difficult because we had to weigh all the castings for each man and keep track of them. To handle them on a day-work basis would have required about 30 instead of 20 men; so, we placed them all in one group and took the foundry report for a certain number of pounds per day, and allowed a certain number of man-hours per 1000 lb. That plan worked well. We dispensed with from six to eight men that had been required under the individual-effort system. The only reason we could get the men to work under this plan was that they had an incentive to increase the output. If we had tried to group them with the molders, for instance, we would not have got anywhere. The plan demands very close co-operation; the men must be near together so they can see that every man's efforts count. A certain output from a group costs us a certain amount of money and, if we are making only one part, the cost of the group operation is the money we pay divided by the number of pieces we obtain. That is positive and actual. We may have various operations going through the group. We may have a heat-treating group in which there are as many as 80 different parts, ranging all the way from small washers to front axles. There may be 20 to 30 men, split into various groups that perform the different operations. We can determine the cost because every one of those pieces has its own standard time. The time in which a front axle goes through is known to be a certain number of minutes or tenths of hour and, if a washer has to be hardened, that would represent a smaller unit of time. When the pay period closes we know certain positive facts. We know the amount of money paid out and the number of pieces produced and we have a standard time-value on each one. We determine the total number of actual man-hours in a group and the total number of standard man-hours and can therefore compute the cost per standard hour. For example, if each standard hour costs us 54 cents and the standard time on a certain product was 0.1 hr., the cost would be $0.054. Above 100 per cent, the cost is uniform. If the efficiency runs down to 90 per cent, the bonus decreases 10 per cent and our costs increase about 1.5 per cent. If the efficiency is reduced to 50 per cent, the cost increases; but, whatever the cost is, it can be computed.

We have very uniform costs. Under the group-bonus plan the cost efficiency is maintained above 90 and 100 per cent fairly well, and we get a uniform cost for given base-rates. No matter how many or how few pieces go through, or how many dividends there are, we can always determine the costs.

The general superintendent of an important factory that has been using the group-bonus system for about 5 months wrote me as follows:

> The results shown already are certainly most gratifying to us and substantiate our belief that the complete installation of this system in our factory will result in a wonderful saving to our company. A rough average of the results disclosed about a 23-per cent reduction in the number of employes, compared with the old method, 20-per cent increased real earnings for employees, and 49-per cent increased man-hour production.
>
> As you are aware, we installed this system a small group at a time, explaining to the employes its advantages to them and to the company, and in no instance have we had any difficulty in throwing-in additional groups. We are frank to say that the company and the employes are sold 100 per cent on this system, and the factory in general is benefited under this new method of working, as it is more quiet and orderly than under the old method. We hope to complete the necessary details and to have all operations working under this system by the first of the year.

To show why these men are taking-up this group-bonus system, I will include an extract from a letter written by the general manager of a company that builds a well-known car. He says:

> After years of experience, I am of the opinion that the group-bonus system is the best system adaptable not only to progressive manufacturing, but to other classes of work where the measurement of individual production would be expensive or burdensome. We are using the group-bonus plan because we feel that it is better adapted to our needs than the former individual-effort system, and that it is much more satisfactory than the usual piece-work system to both the management and our employes. Our experience during the past year has confirmed our views on this subject.

Another extract from an unsolicited letter by the general superintendent and the chief executive officer in one of the largest automobile factories in this Country is as follows:

> In regard to the group-bonus wage-incentive system, one of the first benefits is a doing away with a large number of clerks and carriers that were necessary under the old method of payment. We have found also that we have less labor trouble. I believe this is because the men take more interest in their work. We have found also that we are getting more production where we have this system thoroughly worked out. Another item of interest is the fact that we have reduced the percentage of scrap.

This last factory was formerly 100 per cent on the individual-effort system and had been ever since its start. It had worked on the individual-effort system for years. It was turned over to a group-bonus plan a few years ago and is still operating that way.

I wish to show that the factory managers who have taken this up and have tried it are with us on the proposition. I have received other letters along the same line. Unless these men were with us and found the group-bonus plan to be the right thing, there would be no use trying to put it through. It succeeds because each one who tackles it right and gives it proper cooperation, becomes one of our good friends and boosters.

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Aircraft-Engine Practice as Applied to Air-Cooled Passenger-Car Engines

By S. D. Heron[1]

Annual Meeting Paper *Illustrated with* Drawings and Photographs

STATING that most of the copying of aircraft practice in post-war car-design has proved a failure because the fundamental difference in duty has not been realized, the author proposes to show wherein the automobile designer and the engine builder can profit by the use of practice developed for air-cooled aircraft engines and, after generalizing on the main considerations involved, discourses on the simplicity of layout of the efficient air-cooled cylinder as a preface to a somewhat detailed discussion regarding cylinder design and performance, inclusive of valve location, type of finning and form of cylinder-head.

Cylinder material, cooling surface, port arrangement for in-line engines, and the camshaft and valve-gear arrangement most desirable are subjects treated at length and illustrated, the thought then passing to a consideration of sleeve-valve types, temperature and exhaust-valve cooling, in such detail as to include many enlightening data; and similar treatment is accorded the subjects of air supply, a cooling system for in-line engines, air-blast direction and the necessary quantity of air. Desirable types of fan, spark-plug, piston and fuel are commented upon somewhat briefly. The author concludes (*a*) that scaling-down of design in internal-combustion-engine cylinders is a safe process, while scaling-up is decidedly unsound; (*b*) that, since thermal troubles have been the greatest cause of difficulty with air-cooled cars, a consideration of the designs capable of continuous full-throttle operation is likely to demonstrate the ease with which the relatively mild thermal difficulties of an air-cooled car can be overcome.

Appendix 1, by C. Fayette Taylor, treats the subject of the experimental development of air-cooled engine-cylinders specifically; it reviews briefly the methods used by the Engineering Division in developing and testing cylinder units for both air and water-cooled engines, and discusses some of the results obtained with several of the air-cooled cylinders mentioned in the paper proper. Performance investigations of experimental cylinders are described and commented upon, inclusive of illustrations and tabular data that are given rather lengthy attention. Laboratory experience in regard to exhaust-valve cooling is related, together with mention of the experimental results with internally cooled valves. Subsequently, many air and water-cooled-type comparisons are made, the conclusion being that there will be a rapid and healthy growth of the air-cooled type of powerplant for automotive vehicles.

Appendix 2, by E. H. Dix, Jr., has as its subject the foundry production of air-cooled cylinders with the fins cast integrally and first describes the experience of the Engineering Division foundry in casting aluminum-alloy cylinder-heads, numerous illustrations of castings being included. The suitable gating for aluminum-alloy casting is discussed, supplementary illustrations being presented, and the conclusion is reached that the experience related tends to show that air-cooled cylinder-head castings of the types illustrated are far from being bad production propositions.

THE aim of this paper is to show wherein the automobile designer and builder can profit by the use of practice developed for air-cooled aircraft engines. The aircraft-engine designer is apt to overemphasize the importance of full-throttle performance in car engines, forgetting that the modern automobile engine is essentially a smoothly rotating mechanism rather than a problem of intense heat-flow. Much of the slavish copying of aircraft practice in post-war car-design has proved a failure, because the fundamental difference in duty has not been realized. Nevertheless, it is felt that much of aircraft practice can be used in car design with good effect. There is no reason why more efficient and more uniform cylinder cooling and increased exhaust-valve durability should in any way reduce silence or smoothness. Rather the reverse is true, as an over-heated engine with distorted cylinders and leaky valves is not usually remarkable for either smoothness or mechanical silence.

The average automobile engineer derides aircraft-engine practice on the ground of assumptions such as that the exhaust-valves have to be discarded after 100 hr. of running, or that the whole engine has only a gross life of about 300 hr. of running. Neither of these assumptions happens to apply to modern engines or even to several 1914 and 1915 designs. It would be of interest to know how many modern car-engines are capable of running 100 hr. at three-quarter throttle for say 4000 miles, and what would be the relative condition of any passenger-car engine after such a test in comparison with an aircraft engine after a similar test. There is little question that the comparison would be altogether favorable to the aircraft engine. The rather grotesque attempts at exhaust-valve cooling evident in most cars, both air and water-cooled, lead to the conclusion that the use of aircraft practice would be productive of great improvement. Aircraft practice in this connection would result in greatly increased durability for equal cost.

The application of a large part of aircraft practice would not involve greater cost than that of present-day car-practice. Complication and expense do not necessarily spell efficiency, and crudity is not necessarily economy. Much of the crudity seen in air-cooled engines is decidedly very expensive.

Air-Cooled Cylinder Layout

Aircraft practice has shown how remarkably similar in layout the best air and water-cooled cylinders are. For both types, the layout with inclined overhead valves can be said to have proved most efficient as regards output, fuel economy and cooling. The successful air-cooled cylinder consists essentially of the body of a first-class water-cooled cylinder with the water-jacket replaced by

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suitable fins. The use of air-cooling does not involve throwing into the discard all of the experience gained in the construction of water-cooled engines, but rather the use of that knowledge with due consideration of the differing physical characteristics of the cooling medium.

The modern air-cooled aircraft-engine cylinder is admittedly expensive and apparently complicated in comparison with automobile-engine practice. It is, however, the necessity for minimum weight and sustained high specific output together with high thermal efficiency that is responsible for both the expensive built-up construction and the use of high-class material. High-duty air-cooled cylinders can be built of commercial materials throughout. Alloy-steel exhaust-valves are needed, but not more so than in present water-cooled types. A plain cast-iron cylinder-body with an aluminum piston and a chromium-steel exhaust-valve can, with suitable design, be made to cool efficiently and pull brake mean-effective pressures of over 100 lb. per sq. in. for prolonged periods with a full-throttle fuel-consumption of less than 0.6 lb. per b.hp-hr.

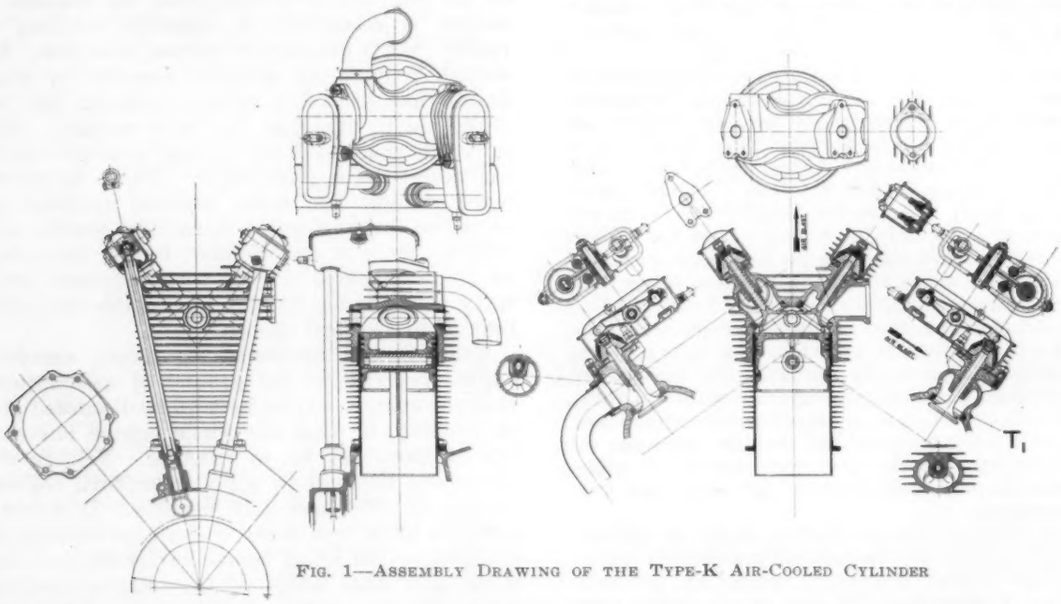

FIG. 1—ASSEMBLY DRAWING OF THE TYPE-K AIR-COOLED CYLINDER

A modern air-cooled aircraft-cylinder is illustrated in Fig. 1. This has a cast aluminum head, screwed and shrunk onto an integrally finned steel-barrel. The valve-seat inserts are of aluminum bronze and are shrunk into the aluminum head, the head metal being peened over a shoulder on the inserts before the head and the seats are finish-machined. While the construction is undoubtedly expensive when measured by automobile standards, the production is really a straightforward proposition. The head casting is a very simple job, being produced with a stripper-plate pattern. It does not involve the use of any cores of loose pieces in the mold, has only a single-body core and, in addition, is cast without the use of chills. While the operations are lengthy, none of them is uncontrollable or uncertain.

This design bears comparison with proposed and existing automobile-engine designs having fins that are cast-in or otherwise attached in such a manner as largely to prohibit inspection of the soundness of attachment.

CYLINDER PERFORMANCE AND DESIGN

Owing to the high friction-loss in the universal-test engines used by the Engineering Division for single-cylinder testing, considerable disparity is evident in the apparent brake performance of different cylinders when they are of widely differing capacity. The author, therefore, in his portion of the paper, will correct the apparent brake performance to a constant-friction mean effective pressure of 15 lb. per sq. in., which is a fair average figure for a modern aircraft engine or a single-cylinder installation designed for minimum friction; brake horsepower, brake mean effective pressure and fuel consumption will be quoted on this basis.

A constant-friction mean effective pressure is really a much sounder basis to use than one of constant mechanical efficiency, since engines of similar construction except as regards cylinders usually have approximately an equal-friction mean effective pressure, although the brake mean effective pressure may vary widely and thus result in a considerable difference in the mechanical efficiency. In few cases is it possible to duplicate a single-cylinder output and fuel-consumption on multi-cylinder engines, and it may seem useless to quote such figures as they are at present generally impossible to attain. Nevertheless, such figures show the potential possibilities of the cylinders on multi-cylinder engines with suitable distribution systems, and indicate what can be and is done in certain cases, such as with radial aircraft engines with rotary mechanical distribution, if the charge be taken into the cylinder under similar conditions to those obtaining in a single-cylinder unit.

Very little reason has arisen for modification of the author's views on the subject of cylinder design during the last year. It will be necessary, however, for the sake of continuity to reiterate some of the ground covered in his last contribution to the Society on Some Aspects of Air-Cooled Cylinder Designs and Development.[3] Recent work has also given ground for a modification of some of the views previously held.

VALVE LOCATION AND TYPE OF FINNING

The overhead valve-cylinder is superior to the side-valve type as regards output, cooling, lack of distortion, fuel economy, ease of production and symmetry. The only advantage of the side-valve design is that of possibly more silent valve operation, and this can be largely overcome by careful design of the overhead-valve operating-mechanism. The outstanding advantage of the overhead

[3] See THE JOURNAL, April, 1922, p. 231.

type lies in cooling. In water-cooled car-engines, the overhead location usually is chosen because of the characteristically greater output, but this is really a minor consideration for an air-cooled car-engine in comparison with the marked cooling advantage. No further consideration will therefore be given to the side-valve type, beyond stating that a first-class side-valve type is much superior to a poorly cooled over-valved overhead-type.

The argument in favor of axial finning on the score that it is productive of uniform air-distribution to all parts of the barrel is basically unsound. The heat given to the cylinder-head and the barrel varies markedly throughout. Since uniform temperature-distribution is desired, it follows that this cannot be secured if the air is uniformly distributed over a surface having varying heat reception. The circumferential fin has the following advantages over the axial fin:

(1) Ease of foundry production

(2) More complete finning and cooling of the cylinder-head is obtainable

(3) A very considerable stiffening effect is obtained, counteracting any tendency of the cylinder barrel to go out-of-round

(4) With a suitable blast distribution, a very uniform distribution of the temperature throughout the head and the barrel can be secured

(5) Dirt is removed more readily from the fin cells. The axial-fin type has closed air-cells which are obtained either by bending the fin-tips or by a closely fitting jacket; both methods result in a long narrow passage from which the removal of burnt-on dust and oil or mud is a matter of some mechanical difficulty. In the circumferential-fin type, any accumulation of foreign matter is apparent on a casual inspection and is easily removed with a wire brush or compressed air

(6) The long narrow parallel air-cells of the axial-fin type are in themselves the cause of a heavy drop of pressure-head in the cooling air. The bad entry and exit from the cells, in conjunction with the sharp right-angle turns at the entry and the exit, are further causes of loss of head. From the resistance viewpoint, the easy convergence and divergence of the air at the entry to and the exit from the cells in the circumferential-fin type present a marked contrast

The normal flat-head air-cooled cylinder with axial finning betrays an appalling crudity in cylinder-head cooling. As a general rule, the head and the ports are completely bare of finning and, in nearly all cases where finning is attempted, the result is worse than a complete absence of fins, as the main results obtained appear to be the disturbance and the obstruction of the air-flow over the cylinder and a marked increase of the casting difficulties with no compensating cooling advantage. It is, however, possible to fin the cylinder crown of an axial-fin type efficiently, as regards both heat abstraction from the crown wall and dissipation to the cooling air, without causing broken or obstructed air-flow over either the crown or the barrel. Figs. 2 and 3 show two such designs by H. E. Morehouse, of the Engineering Division. These have evidence of original attack and great care, and are the first designs seen by the author that show any real attempt at head cooling of an axial-fin type. The author's admiration for these designs, however, in no way alters his belief in the superiority of the circumferential-fin type.

As regards evenness of temperature distribution in

[3] See THE JOURNAL, April, 1922, p. 231.

[4] See THE JOURNAL, April, 1922, p. 231.

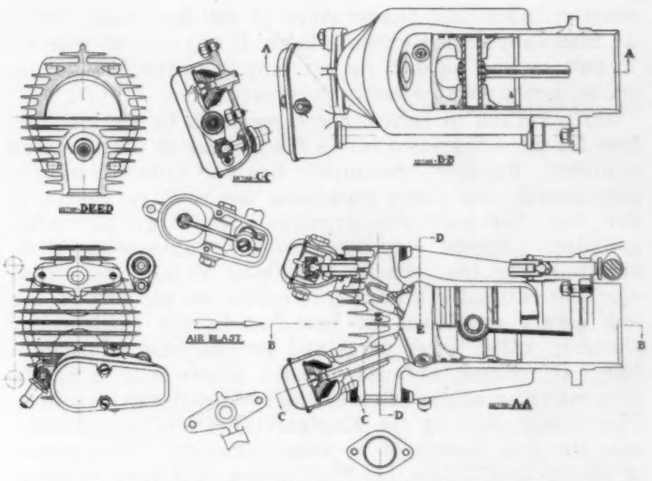

FIG. 2—THE MOREHOUSE AXIAL-FIN CYLINDER WITH HORIZONTAL PORT OUTLETS

the circumferential-fin type, the B. S. A. car-engine cylinder, with exhaust-side blast, illustrated in Fig. 4, showed a maximum circumferential temperature-difference at full throttle of 27 deg. fahr. at the top of the barrel and 47 deg. fahr. at the base of the barrel.[3]

FORM OF CYLINDER-HEAD

Recent work has confirmed the author's views still further as to the fundamental fault of the flat-head type of cylinder, and comparative tests of the types I and J Engineering-Division cylinders indicate sharply the inferiority of the flat design. Both types are of 5⅝-in. bore and 6½-in. stroke; type-I has a flat head with twin exhaust-valves and a single inlet, and type-J a hemispherical head with single inlet and exhaust-valves. These cylinders give a very fair comparison as both are by the same designer, were produced during the same period and have a common construction,[4] except for the number of valves and form of cylinder-head. The type-I, with internally cooled exhaust-valves, developed a maximum brake mean effective pressure of 132 lb. per sq. in. on a fuel-consumption of 0.62 lb. per b. hp-hr., with a maximum cylinder-head temperature of 570 deg. fahr. The type-J, with a normal exhaust-valve, developed more than 137 lb. per sq. in. brake mean effective pressure on a fuel-consumption of 0.48 lb. per b. hp-hr. with a maxi-

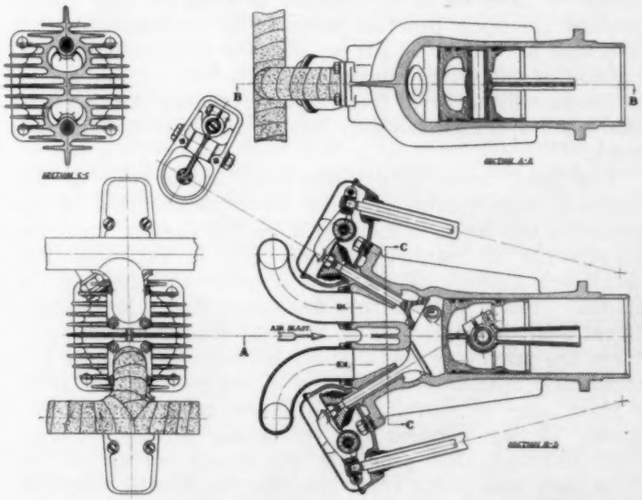

FIG. 3—THE MOREHOUSE AXIAL-FIN CYLINDER WITH VERTICAL PORT OUTLETS

mum cylinder-head temperature of 450 deg. fahr. With an internally cooled exhaust-valve, it has proved possible to run as lean as 0.43 lb. per b.hp-hr. with 123 lb. per sq. in. brake mean effective pressure.

With regard to valves, there seems to be no justification for more than two valves for any car or truck-engine cylinder. Recently, the author has had cause to modify considerably his views regarding the relative merits of two and four-valve constructions for large air-cooled cylinders. Recent experience of the Engineering Division indicates that, for up to at least 50 b.hp. per cylinder, the two-valve type is preferable on account of the less severe conditions of heat-flux in the combustion-chamber walls, fewer parts and general ease of production. Of course, the use of such power with a single exhaust-valve necessitates efficient exhaust-valve cooling. The recent work of the Engineering Division indicates that this is a matter of no great difficulty. The success of the 91 and 122-cu. in. Fiat racing cars goes to show that multi-valves are not a fundamental necessity for a maximum output, even at abnormally high speeds, and that the use of multi-valves for small cylinders in racing cars has been a matter of fashion rather than necessity.

The futility of over-valving has been demonstrated by aircraft engines of both the air and the water-cooled types. If aircraft engines, where the last pound of output is required, can secure the desired result with relatively small valves, obviously, there is no reason for the use of excessive valve-capacity in car engines where the output must of necessity be sacrificed to smooth running, acceleration and even firing at all speeds. The use of large valve-area is required only for abnormal speeds and, even in this connection, has its limits on account of the low turbulence that produces a low rate of flame propagation. For normal purposes, large valves result in a bad combustion-chamber form and an accentuation of the cylinder-head and valve-cooling difficulties.

Automobile engineers generally, and particularly those of European persuasion, seem to consider that the last step in obtaining high output has been taken by the use of large valves, apparently considering that poor torque at the lower speeds, cracked valve-seats in the cylinders and valve-burning produced by a few hours at anything approaching full throttle are necessary adjuncts to high output. Such absurdities as exhaust-valves larger than the intakes should be extinct in these days, yet they are nevertheless to be found. Even if the experimental evidence be ignored, it is difficult to imagine any logical ground for such practice when it is considered that a pressure up to 100 lb. per sq. in. is available for the expulsion of the burnt charge and only 14.7 lb. per sq. in. to produce filling with fresh charge. Therefore, it seems that such a design can only be the result of pure guesswork.

The ideal form of cylinder-head appears to be the complete hemisphere, and the more nearly this is approached the more efficient will be the design from the standpoints of minimum stress due to explosion and maximum amount of fin area directly attached to the cylinder crown. The author has found that a large angle of valve-tilt is productive of the best results, as this produces the maximum metal space between valve-ports and apparently maximum filling and discharge capacity from a given size of valve. An included angle of approximately 70 deg. between the valves has proved satisfactory, although this prohibits the direct boring of the valve-seats and ports from the cylinder mouth when an integral head and barrel are used.

The term detachable cylinder-head is used in the sense of a removable head, and not to represent the separate but non-removable head of the type shown in Fig. 1. While there is little question that air-cooled cylinders with detachable heads can be made to function with fair satisfaction, there is not much doubt that the cooling and general efficiency of the type is lower than that of the solid-head construction. Some of the reasons for avoiding detachable heads are as follows:

(1) The heat given to the head has to be almost entirely dissipated by it, as the transmission to the barrel and its finning is limited owing to the very high thermal resistance of the joint and the gasket. Hence, the head runs hotter and the barrel finning does not carry as much of the cooling duty as it might carry otherwise

(2) Gas leakage at the joint, with the attendant high temperature, is likely to occur, as the pulling-up of a gasket on a single-cylinder head is a matter of considerable skill and requires more than one adjustment

(3) The bolts or studs and their bosses for holding down the head usually cut-up the head finning considerably and break-up the uniformity of air-flow over the head

(4) The uneven sections resulting from the attachment bosses are liable to produce distortion under temperature, in addition to the mechanical distortion that is induced so readily by uneven bolting-down or lack of stiffness in the head or barrel

(5) It is easier to lift a separate cylinder for decarbonizing than to break and remake a head gasket-joint, particularly in a push-rod engine involving the readjustment of the tappet clearance after refitting the head and at intervals while the head joint is settling, as it practically always does

Satisfactory block-type cylinder-construction with air-cooling has yet to be produced and, if the detachable head be assumed to be unsatisfactory, no argument except stiffness of the engine as a whole can be adduced in its favor, as it presents an undesirable foundry problem and is more unwieldy to handle in both manufacture and service. The use of separate-cylinder construction in a water-cooled engine involves lengthening the engine by the thickness of two jacket-walls, the external space between them and the two water spaces, which, together, are usually greater than would be allowed between the barrels with block construction. These disadvantages do not apply in an air-cooled engine, as a certain minimum space for air-flow has to be allowed, irrespective of whether block or separate-cylinder construction is used.

Cylinder Material

The cast-iron cylinder with the fins cast integrally still appears to be the most logical solution for car engines up to the $2,000 class. Recent tests by the Engineering Division, which will later be dealt with more fully, have shown that, in capacities of up to 87½ cu. in. per cylinder, cast-iron cylinders at continuous full-throttle will give cooling, output and fuel economy that are adequate for automobile use.

There is a tendency to assume that some particular material can be found that will solve the difficulties of air-cooling. Design is the dominant factor in air-cooling and no material, whatever its thermal conductivity is, can counterbalance the effect of poor design. It may be well to emphasize that there is no mystery concerning air-cooling. Successful cylinder design is mainly a combination of common-sense and applied science, with very little computation involved. However, the author recommends cast iron as a cylinder material only provided a

suitable design be used. The B. S. A. car-engine cylinder-design illustrated in Fig. 4, and that of the type-K cylinder shown in Fig. 1, represent types that have proved satisfactory.

The designs already mentioned no doubt will be objected to at once for car use, on the score of the hemispherical head. This type of head unquestionably gives rise to a minor machining difficulty, but the elimination of this difficulty by the use of a flat head is sure to result in a final increase of the cost to secure a similar performance, because larger fan capacity, and probably more cowling and more expensive cylinder and valve material, will be required. While the matter has not been the subject of an exact comparative test, it is the author's considered opinion that, from the performance standpoint, the flat-head aluminum cylinder using, except for the flat head, the best of aircraft practice, is in no way superior to the hemispherical type of cast-iron head; in fact, up to 100-cu. in. capacity, it is actually inferior.

If, in the production of a medium-priced air-cooled car-engine, design be subordinated to production, the result is sure to be unsatisfactory. The air-cooled engine has yet to gain the confidence of the entire public, and it is urged that nothing be done to destroy the existing amount of faith by allowing full sway to the production engineer who can see only a production proposition in an air-cooled engine and is entirely uninterested in its functioning as a prime-mover. The Liberty aircraft-engine cylinder is an excellent example of what the production engineer can do if given a fixed piece to produce, without license to obliterate the original design to facilitate production.

It has been found that fairly thick metal is necessary in the walls and fins of a successful cast-iron cylinder. The old idea of thin walls being most efficient on account of the heat having less distance to flow through is altogether unsound in the light of present-day experience. In fact, there is considerable reason to believe that the superiority of aluminum to cast iron for large cylinders is as much due to the low resistance to heat-flow around rather than through the walls (that is, the rapid conduction of heat from the points of maximum heat-reception to the parts having a lower reception) as it is to the greater heat-dissipating powers of the fins. For a 3½-in. bore hemispherical-head cast-iron cylinder, a thickness of ⅝ in. at the center of the combustion-chamber, tapering to 7/16 in. at the edge of the chamber and to not less than ¼ in. at the lowest fin on the barrel, appears to be suitable.

Fairly thick fins are necessary to insure sufficient heat-flow capacity in average castings, ease of production and strength to resist rough handling; the last-named is by no means an unimportant factor. From these considerations, suitable fin proportions appear to be a maximum length of 1 in., a pitch not less than ⅜ in., a thickness at the root of 9/64 in. and a thickness at the tip of 5/64 in. Such fins are produced readily and commercially from stripper-plate patterns with jolt-ramming. To insure clean and sharp casting of fins having the proportions given, it appears to be necessary to have a minimum of 0.5 per cent of phosphorus in the iron.

While the straight cast-iron cylinder has a considerable advantage over more complicated and expensive and at the same time less efficient constructions commonly considered to be essential for air-cooling, it is nevertheless inferior as regards cooling, fuel economy, tendency to detonation, weight and, to a very slight extent, output, to the type of aluminum cylinder used for aircraft

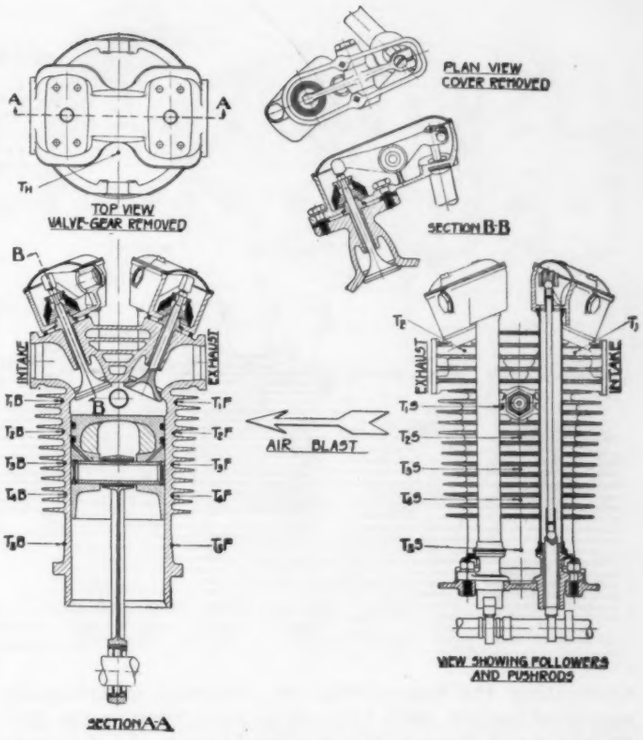

FIG. 4—PLAN VIEW AND SECTIONAL ELEVATIONS OF THE CYLINDER USED IN THE B. S. A. PASSENGER CAR

engines. Where considerations of cost will allow, the use of aluminum cylinders should be given very serious thought. As a basis of comparison with the straight cast-iron type, the general construction used by the Engineering Division will be used. The screwed-on head aluminum-construction is the only type considered as, in general, this has proved the best all-round design for heavy duty, is the easiest type to produce and gives the least trouble with a variation of the thermal contact and the like in a quantity of cylinders. The pros and cons of the various types of aluminum-cylinder construction have been discussed at length previously.[5] The construction is well exemplified by the type-K cylinder shown in Fig. 1. The aluminum-head casting will cost more than the cast-iron, but will be much more easily machined, particularly if valves tilted to a large included angle, 70 deg. in this case, be used. The hemisphere can be machined at a much higher rate of speed, on account of cutting facility and also accessibility, as the hemispherical surface in the case of the aluminum head is not at the bottom of a deep hole. To offset the more rapid and accessible working of the interior of the head, there is the extra work due to the threading for barrel attachment, boring valve-seat insert-recesses and machining and shrinking-in the seat inserts. The seat-insert material also has to be provided; in this case it is extruded aluminum-bronze. This can be obtained extruded in tubular form, very close to the finished dimensions. While the cost of this is not negligible, a remarkably durable valve-seat is obtained and, furthermore, the inserts can be replaced with ease if damaged or worn-out, although the life is probably far in excess of that of any iron or steel seat. The aluminum head will accumulate less carbon-deposit, and the deposit obtained is more readily removable than that on cast iron. Aircraft practice at present uses forged-steel barrels with machined integral fins. The Engineering Division is, however, in-

[5] See THE JOURNAL, April, 1922, p. 231.

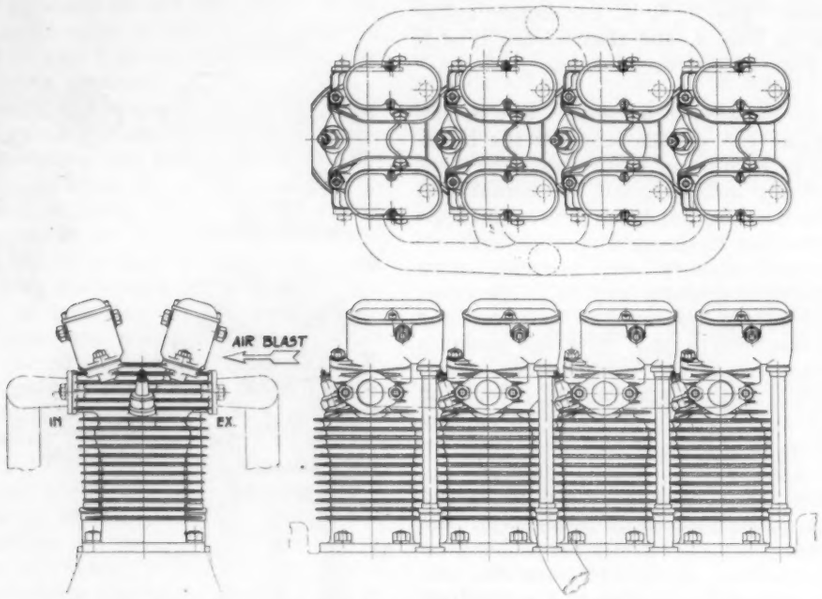

FIG. 6—PLAN VIEW AND SIDE ELEVATION OF A FOUR-CYLINDER IN-LINE ENGINE WITH THE SAME ARRANGEMENT OF PORTS AS THE ENGINE ILLUSTRATED IN FIG. 4

vestigating the possibilities of obtaining centrifugally cast steel-barrels, with little or no machining on the fins. Should this prove possible, use might be made of barrels produced in this manner for car practice. In any case, centrifugal cast-iron barrels appear to be a commercial product for car practice, as centrifugal casting produces a high-grade material having great strength, close grain, excellent wearing properties and uniform machining characteristics, with little wastage due to unsoundness. Threading the head and the barrel should not prove very costly with suitable tooling. The limits need not be excessively fine if the minimum shrink be such as to produce a slight permanent-stretch in the head. If the maximum shrink within the limits set be obtained, this will merely cause an increased stretch of the head. The general result will be to obtain a nearly uniform amount of actual shrink between the head and the barrel, irrespective of the initial shrink. The use of a separate barrel allows the replacement of this part in the event of scoring or other damage. The use of the aluminum head also causes additional expense, due to the necessity for spark-plug bushings.

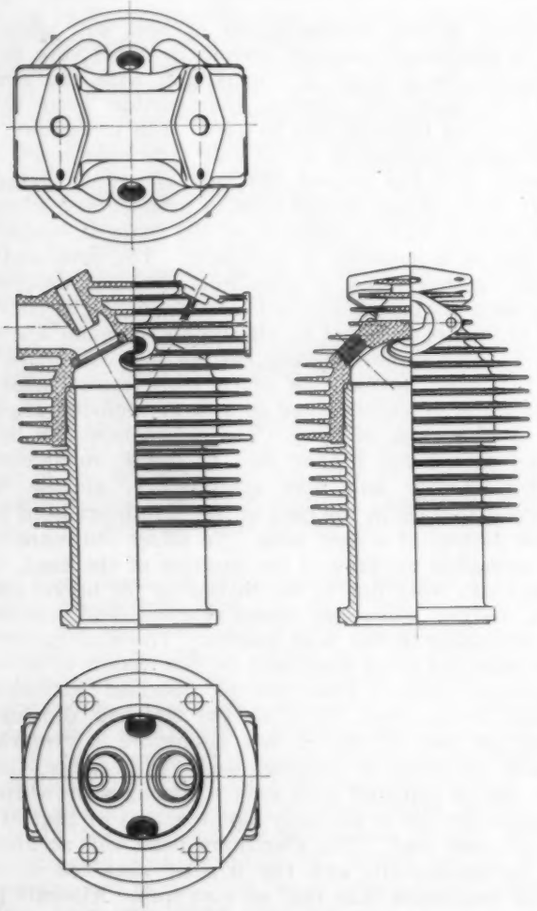

FIG. 5—A MOTORCYCLE CYLINDER WITH AN ALUMINUM HEAD SCREWED ON TO A CAST-IRON BARREL

Fig. 5 shows a successful motorcycle cylinder having an aluminum head with a screwed and shrunk cast-iron barrel. With aluminum heads, rough handling is liable to result in unscrewing the studs from the casting while removing nuts. Replacement of the valve-guides is also liable to result in torn and scored bosses, unless carried out with care. Hard alloys and the use of bronze studs go far toward minimizing the above-named troubles. In any case, the aluminum head does not suffer from the burning together of cylinder metal and studs or valve-guides that is liable to occur with cast iron.

The fin proportions recommended for cast iron are also suitable for use with aluminum. To stand rough handling, finning for car engines should generally be heavier than that used for aircraft engines, the slight increase of weight and possibly of cost being of no moment.

COOLING SURFACE

The cooling surface obtainable with finning of the proportions recommended will not vary very much. The B. S. A. cylinder of approximately 3½-in. bore and 3⅜-in. stroke has 4.18 sq. ft. of area in the fin, head, barrel and port surfaces and, at the normal maximum output of 8.5 b.hp. per cylinder, this is equivalent to almost 0.5

sq. ft. per maximum hp. The larger the cylinder is, the lower the cooling surface obtainable per unit capacity becomes and, while the smaller cylinder usually works at a higher number of revolutions per minute and has higher output per unit capacity, it nevertheless usually has greater cooling surface per maximum brake-horsepower.

The type-K cylinder has 6.68 sq. ft. of cooling surface; the head has 3.60 sq. ft. and the barrel has 3.08 sq. ft. At 110-lb. per sq. in. brake mean effective pressure and a speed of 2000 r.p.m., or 24.3 b.hp., this is equivalent to 0.275 sq. ft. per b.hp. A much higher output is obtainable from this cylinder with a cast-iron head, with satisfactory operation, although a blast velocity of 60 m.p.h. or more has to be used.

The type-J Engineering-Division cylinder has 11.40 sq. ft. of cooling surface; the head has 6.95 sq. ft. and the barrel has 4.45 sq. ft. It has a normal power of 47.5 b.hp. at 1700 r.p.m., although considerably more can be obtained without overheating; this is equivalent to 0.240

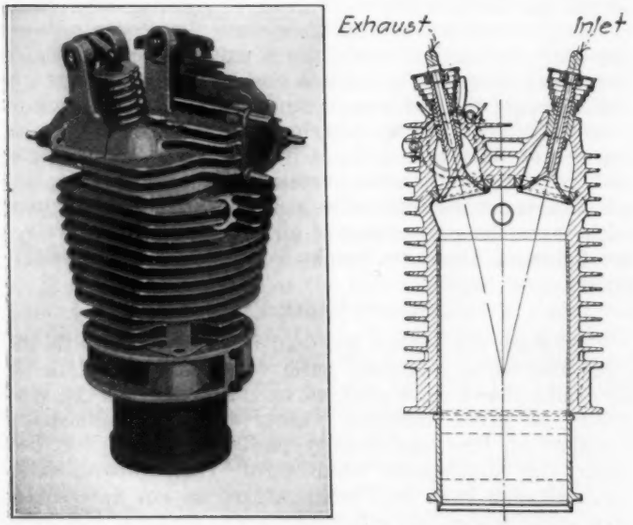

FIG. 7—EXTERIOR VIEW AND SECTIONAL ELEVATION OF THE 4D CYLINDER DEVELOPED BY THE ROYAL AIRCRAFT ESTABLISHMENT

sq. ft. per b.hp. This cylinder cools excellently at full throttle in a 70-m.p.h. blast.

The performance of the B. S. A. engine is of interest in showing the ease with which air-cooled cars with reasonable cylinder design can be cooled. This engine, of only 65-cu. in. capacity, gives satisfactory cooling on the road while pulling a car weighing 1700 lb. when loaded. It has no cooling fan, the motion of the car being relied on entirely for the cooling-air supply. An engine of such a capacity works at a considerably higher mean power-factor than do engines of capacities suitable for use in this Country. In view of this performance, the author has no hesitation in claiming that extremely efficient cooling can be obtained from any four or six-cylinder car-engine having finning of the recommended proportions and a cooling-surface allowance of 0.4 sq. ft. per maximum b.hp. This allowance, in conjunction with a suitable cooling fan that absorbs not over 3 per cent of the maximum engine brake horsepower, should give excellent cooling at continuous full-throttle on the road, provided reasonable cylinder-design and air-distribution be used.

PORT-ARRANGEMENT FOR IN-LINE ENGINES

It is of interest to examine the possible port-arrangements with hemispherical-type cylinder-heads on in-line

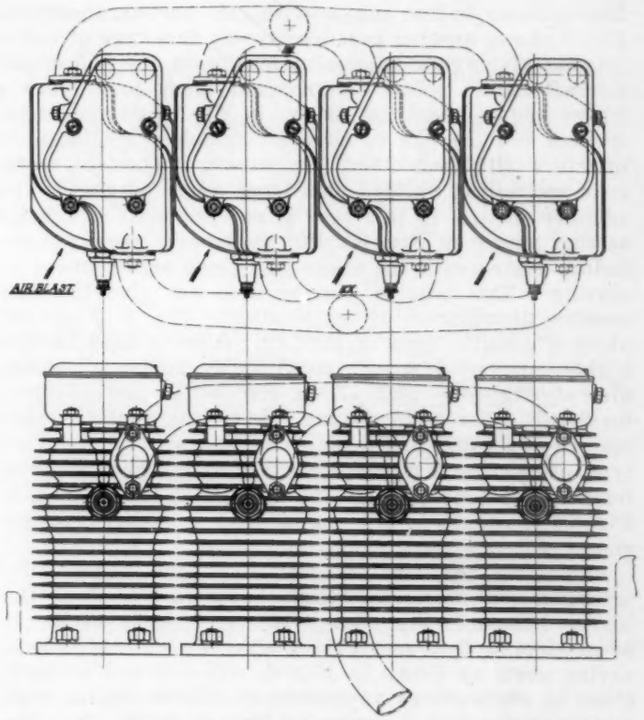

FIG. 8—PLAN VIEW AND SIDE ELEVATION OF A FOUR-CYLINDER IN-LINE ENGINE WITH THE SAME ARRANGEMENT OF PORTS AS THE ENGINE ILLUSTRATED IN FIG. 7

engines. Fig. 4 shows what is probably the best arrangement, as it allows exhaust-side draft. This type gives the most uniform cooling and results in the most readily molded cylinder casting. Fig. 6 shows a layout of a

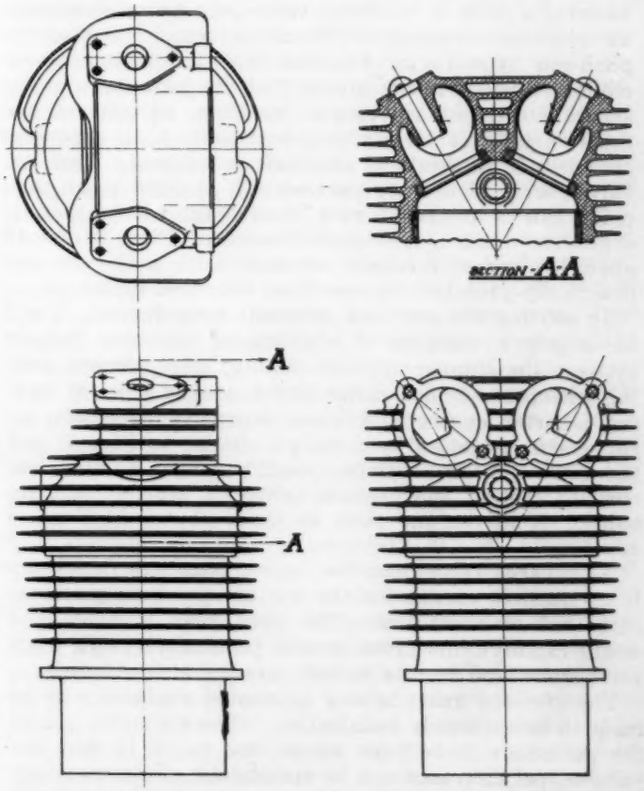

FIG. 9—PLAN VIEW, ELEVATIONS AND SECTION OF A HEMISPHERICAL-HEAD CYLINDER WITH BOTH PORT OUTLETS ON ONE SIDE

four-cylinder in-line engine with this port-arrangement. Fig. 7 shows another possible layout; this type of cylinder has given very successful results on the R.A.E. 4D aircraft-engine. This arrangement tends to produce a longer engine than that shown in Fig. 6, as the valve-springs and the like of adjacent cylinders are liable to interfere with each other, particularly if enclosed valve-gear and valves inclined at a large angle be used. The cylinder casting in this case is not produced so readily as that shown in Fig. 6. Fig. 8 shows a four-cylinder in-line engine with the above-mentioned arrangement of porting. This type of porting does not give the best possible distribution of temperature. Fig. 9 shows yet another possible type of porting. This is used on the highly successful Armstrong-Siddeley radial air-cooled aircraft-engines. This allows the intake and exhaust-manifolds to be on the same side of the engine, which may be of advantage for hot-spotting and stoving. This type of porting has the same foundry difficulties and temperature-distribution disadvantages as that of Fig. 7. Fig. 10 shows an in-line engine with this port-arrangement.

Regarding the relative merits of the three port-arrangements illustrated, that shown in Fig. 6 is the best and those in Figs. 8 and 10 are of equal merit. It is considered that an engine with cast-iron cylinders, having ports as shown in Fig. 6, will cool and perform about as efficiently as an aluminum-cylinder engine with porting such as that shown in Figs. 8 or 10. For the best all-round design, there is little question that an aluminum engine with Fig. 6 porting is superior, as this will allow large valve inclination, permit the shortest engine and give the most efficient cooling.

Camshaft and Valve-Gear Arrangement

The port-arrangement shown in Fig. 6 necessitates two camshafts with a push-rod valve-gear or, alternatively, an overhead camshaft. The author considers that the push-rod type is by far the best for a touring-car engine, when it is considered that the advantage of the separate cylinder as regards handling, adjustment and decarbonizing is largely lost with solid-head air-cooled cylinders if an overhead camshaft be adopted. Push-rod valve-gears, to operate successfully at 3000 r.p.m. and more, can be produced with careful but not necessarily expensive design. The port layouts of Figs. 8 and 10 allow the use of a single camshaft with push-rods and practically prohibit the use of an overhead camshaft.

In settling the port and camshaft arrangement, it will be largely a question of striking an economic balance between the simpler cylinder-casting, more efficient cooling, cheaper cylinder material for equally efficient cooling, shorter engine and more expensive camshaft arrangement produced by a design similar to Fig. 6; and the more difficult castings, possibly longer engine, less efficient cooling and cheaper camshaft arrangement resulting from designs such as those shown in Figs. 8 and 10.

As regards valve-gear, the author considers that there is no possible excuse for the use of open-type gear, except that of cheapness. The open gear is noisy and wears rapidly; this gives trouble particularly with worn valve-stems and results in bad running at low speeds.

The enclosed gear, having lubricated surfaces, can be made to be extremely satisfactory. The enclosure causes the push-rods to be kept warm; the result is that the valve-tappet clearance can be maintained almost constant by the use of suitable push-rod material. An enclosed valve-gear, when using suitable push-rod material (such as aluminum, bronze or steel, depending upon the type and the material of the cylinder and the temperature attained by the rods), backed-off cams and positive lubrication, can be made remarkably silent in spite of the greater resonance of an air-cooled cylinder in comparison with a water-cooled cylinder.

The type of enclosed valve-gear shown in Figs. 1 and 4 has, in the author's experience, given considerable satisfaction. While the angles of the boxes, push-rods and rockers appear to result in complication, nevertheless the parts are really remarkably easy to produce and the difficulty of the type is mainly confined to the design office.

The lubrication of the type of valve-gear shown is satisfactorily attained by breathing through the valve-gear, vertical slots being cut in the cam-follower guides and the vapor being exhausted through a small hole in each rocker-box cover. In the B. S. A. 90-deg. V-type engine very satisfactory lubrication was obtained by breathing through the valve-gear, this being assisted by the large variation of crankcase pressure occurring during every revolution, producing a pulsation of vapor into the boxes from the crankcase and back again. Breathing through the valve-gear tends to produce an engine with an oily and dirty exterior, and allows the possible ingress of dust and grit. Direct oil-feed to the valve-gear is probably the most certain system and has decided advantages as regards valve-cooling, but it must be used with caution, as an excess of oil will result in the valves not following the cams, smoky exhaust and fouled spark-plugs.

Sleeve-Valve Types

Although the author has had no experience with the type, and while relatively little work has been done on air-cooled sleeve-valve engines, he considers that the type should be given thought. Tests by the Engineering Division on the cast-iron type-K cylinder, described under the subsequent heading of Temperature, have indicated that high wall-temperatures do not necessarily have such deleterious effects as are commonly assumed. It seems that the unquestionably high internal wall-temperatures of the sleeve-valve engine may be of no serious disadvantage, especially in view of the absence of red-hot exhaust-valves.

Temperature

The author has discussed previously, at some length, the effect of mixture-strength, compression-ratio and blast velocity and direction upon the temperature and the output.[6] Since then, however, reason for sharp modification of the views expressed has arisen.

Tests on a cast-iron head type-K cylinder such as is shown in Fig. 1 have been the cause for the modification of viewpoint. The cylinder in question is exactly as shown in Fig. 1, except that the bronze valve-seat and spark-plug inserts are omitted, and it has a cast-iron head screwed and shrunk onto the integrally finned steel-barrel. The cylinder is of 4½-in. bore and 5½-in. stroke; it has an 87.5-cu. in. capacity and an internally cooled wide-seat tulip exhaust-valve, the valve-gear being lubricated by forced feed. The best performance is a brake mean effective pressure of 140 lb. per sq. in. at 1800 r.p.m., or 28.0 hp., with a 5.3 compression-ratio and an 0.56-lb. fuel-consumption when using a mixture of 80 per cent of aviation gasoline and 20 per cent of benzol. This performance was secured with an air-temperature of 55 deg. fahr. and a head-temperature of 730 deg. fahr. at T_1, the blast speed being 100 m.p.h. It is possible

[6] See The Journal, April, 1922, p. 231.

under these conditions to run continuously on full throttle with no thermal trouble and no more detonation than is usual with an aircraft engine of high output. Not unnaturally, a cylinder having such high wall-temperatures is rather sensitive to such conditions as air-temperature and spark-advance, and it has not proved possible to duplicate regularly the output just quoted. However, it has been found possible to maintain regularly at least a brake mean effective pressure of 130 lb. per sq. in. at 1800 or 2000 r.p.m., with head-temperatures up to 800 deg. fahr. In this cylinder, the last 10 lb. of brake mean effective pressure is obtained at the cost of a very considerable increase of the head-temperature, the difference between 120 and 130 lb. causing an increase of temperature of about 150 deg. fahr.

The best performance of this cylinder was obtained under nearly ideal conditions, as an extremely large carbureter was used to secure the maximum volumetric efficiency and the maximum temperature of the exhaust-valve was about 900 deg. fahr., this temperature being confined to a very small zone. It was thought at first that the high output was due to the internally cooled exhaust-valve, eliminating the red-hot surface within the cylinder; and that, under such conditions, high combustion-chamber temperatures were of no great moment. This assumption, however, was proved incorrect by comparative tests of cooled and uncooled valves. The tests were carried out on the same day. With the cooled valve, a brake mean effective pressure of approximately 135 lb. per sq. in. was obtained; the maximum temperature of the valve-head was approximately 900 deg. fahr. and the stem was dead-black. With the uncooled valve, no change in the power or the cylinder-head temperature was evident. The uncooled valve attained a maximum temperature of about 1300 deg. fahr., which is decidedly hot. In view of the fact that the cylinder-head temperature was more than 700 deg. fahr. and that the valve was not of the best design for use without internal cooling, it was a cause of surprise that the valve-cooling was as efficient as it proved to be, especially as similar valves in many water-cooled engines run just as hot, although with cylinder-head temperatures that are probably of the order of 400 deg. fahr.

In the author's experience, cylinder-wall temperatures of the order of those attained by the type-K have hitherto invariably been associated with low output, excessive thermal trouble with the valves and the cylinder-head, detonation and a very low thermal efficiency. The type-K cast-iron cylinder has had more than 50 hr. of rather brutal full-throttle testing and, during this time, it has developed no thermal trouble beyond burnt or disintegrated spark-plugs. This cylinder attains wall-temperatures that militate against fuel economy, and the minimum fuel-consumption obtained was about 0.45 lb. per b. hp-hr., this consumption being obtained only by very delicate adjustment; it could be duplicated or bettered with much greater ease in an aluminum cylinder.

The blast direction shown in Fig. 1 is not the most suitable, and some reduction of the temperature could be obtained by the use of exhaust-side blast in conjunction with a port-arrangement similar to that in Fig. 4. The cast-iron type-K cylinder can be operated at full throttle in a blast velocity as low as 60 m.p.h. without any reduction of the output or an increase of the fuel consumption, although, of course, the head-temperature and the tendency to detonation increase.

There is no question that the cooling of this cylinder would be improved markedly if a heavy integral cast-iron barrel were used in place of the light screwed-in steel-barrel. When choked down to a brake mean effective pressure of 105 or 110 lb. per sq. in. and arranged with a suitable blast direction and port layout, a cast-iron cylinder of even this size should give very satisfactory results in car or truck service, although such power per cylinder is hardly likely to be required. When throttled to a brake mean effective pressure of 110 lb. per sq. in., a considerable reduction in the head-temperature can be expected and excellent fuel-economy should be obtained without difficulty.

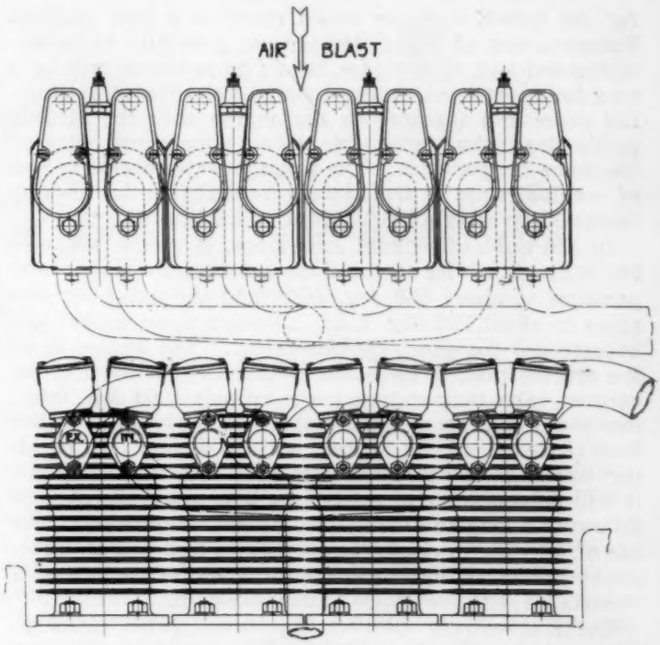

FIG. 10—PLAN VIEW AND SIDE ELEVATIONS OF A FOUR-CYLINDER IN-LINE ENGINE WITH BOTH PORT OUTLETS ON THE SAME SIDE

The fuel economy between three-quarter and full throttle is usually of little importance on a car or truck engine, as relatively little running is done in this range. As the effect of an increased specific fuel-consumption is to reduce the cylinder temperature markedly it is probable that beneficial results will be obtained by progressively stepping-up the consumption from 0.55 lb. at three-quarter throttle to 0.70 or 0.75 lb. at full throttle. This will tend to secure the maintenance of a constant cylinder-temperature and the elimination of detonation, and probably will not increase the fuel consumption in miles per gallon on account of permitting more full-throttle work and requiring a lower power-consumption in the cooling fan.

It is commonly said that air-cooled car-engines pick-up poorly on high-gear from low speeds. This is probably caused by charge-heating due to the hot walls, and the relatively large period available for heating at low speeds. In the author's limited experience with air-cooled car-engines, however, poor pick-up at low speeds has not proved to be a necessary evil of air-cooling. Torque at low speeds and pick-up are closely connected, and comparatively the most determined puller at low speed on high-gear that the author has had experience with was an air-cooled two-cylinder car-engine. A very probable cause for poor pick-up on air-cooled cars appears to be the very odd heavily overlapped valve-timing that is considered necessary. Such timing, while desirable for aircraft or racing engines, is not by any means suitable for a touring-car engine.

It is probable that such temperatures as are quoted

for the type-K cylinder would result in a poor pick-up. Temperatures of this order are not generally to be recommended and, in any case, would be produced only by a long full-throttle pull. They could be eliminated by shifting gears and speeding-up the engine with the throttle partly closed thus increasing the air supply and reducing the flame temperature. The average water-cooled engine of medium class is not usually remarkable for pick-up immediately after a 15 to 20-in. pull on full throttle.

In the light of present knowledge, it seems desirable for average service to limit the maximum cylinder-temperature to about 600 deg. fahr. and the mean temperature to about 350 deg. fahr., although these figures can be exceeded for short periods safely. The author is of the opinion that, if measures be taken to insure that the exhaust-valve temperature never exceeds 1000 deg. fahr., maximum and mean temperatures considerably higher than those quoted can be used without danger. In deciding how high the temperatures are to be allowed to go, it will be necessary to settle just how much the pick-up following a long full-throttle burst can be sacrificed. The use of very high cylinder-temperatures also has an undesirable influence on detonation, a factor of considerable importance with low-grade automobile-fuel.

The Engineering Division has investigated temperature conditions in the wall of a Liberty-engine cylinder and, although this investigation is still in progress, the preliminary results are of great interest for comparative purposes. The tests were conducted with full throttle at normal speed, water temperature and water circulation. With walls and jackets free of scale, the maximum temperature in the combustion-chamber wall was 310 deg. fahr. When the walls had accumulated an 0.025-in. thickness of scale, which is approximately the maximum for a Liberty-engine cylinder because flaking-off occurs on reaching this thickness owing to the deflection of the head walls, the maximum temperature rose to 430 deg. fahr. Heavily scaled cylinder and jacket-walls are the normal condition of the average water-cooled engine, scale of even ⅛ or 3/16-in. thickness being not unusual. The effect of such a coating on the wall temperature can be imagined readily if 0.025 in. is sufficient to account for a temperature-rise of 120 deg. fahr. in a Liberty engine cylinder.

The Liberty-engine tests indicate that there is probably little difference in the cylinder-wall temperature of a good air-cooled cylinder and that of a water-cooled type after it has seen some service and become scaled-up. Large aluminum aircraft-cylinders, such as the Engineering Division type-J that develops well over 40 b.hp., will run on continuous full-throttle for 8-hr. periods with maximum cylinder-head temperatures of 400 to 450 deg. fahr.

Exhaust-Valve Cooling

The design of the exhaust-valve port and seat in the average car engine, whether water or air-cooled, is extremely crude from the viewpoint of the proponent of the aircraft engine. A run of 10 hr. with full throttle at 2000 r.p.m. will work severe havoc with the exhaust-valves and seats of most car engines; whereas, in an aircraft engine, the valves will merely be nicely run-in after such a test.

Much of the barbaric practice exhibited in car-engine exhaust-valve cooling has not even the virtue of being cheaper than good design. The worst features of average practice are as follows:

(1) The width of the seat in the cylinder is usually extremely small
(2) The rim of the valve is often very thin
(3) A considerable length of the valve-stem is often directly exposed to flame
(4) The length of the valve-guide within the exhaust-port is usually directly exposed to flame, instead of being shrouded by the valve-guide boss
(5) Valve-stems of very small diameter are often used
(6) Valve-seats in water-cooled engines are rarely in direct contact with water for the entire circumference of the seat. In many air-cooled engines no attempt is made to circulate air between the intake and exhaust-ports. On the contrary, a wall of solid metal usually joins the two ports at the point of closest proximity

Taking these points in order, the disadvantages of each will be enumerated. The narrow seat referred to in (1) involves a high specific rate of heat-flow from the valve to the cylinder seat. This increases the temperature-drop from the valve to the cylinder and thus increases the valve temperature. Apparently, narrow seats are used for their rapid-seating qualities although, generally, they seem to unseat almost as quickly as they seat. Extremely wide seats have a tendency to pit slightly in excessively oily engines, on account of carbon being hammered into the face of the seat of either the valve or the cylinder, whichever be the softer.

The bad qualities of valves with ultra-thin rims, mentioned in (2), are not very readily apparent on the surface. However, consider the thermal conditions in a valve that is blowing-by in the seat, a condition that will occur more or less in service. Hot gas is blown through an aperture of small sectional-area and large exposed surface, which removes most of the available heat-content of the gas. The gas is at a high temperature and pressure and possesses an extreme cutting-quality on red-hot metal. When blowing occurs, a large volume of heat is liberated in the leak aperture. In the cylinder, this heat will flow radially, circumferentially and axially in the cylinder metal away from the location of the blow. In the valve, the heat will flow circumferentially and radially to the remainder of the valve-head. If the valve has a thin rim, there will not be sufficient heat-flow capacity to conduct the heat to the cooler portions of the valve fast enough to prevent the metal at the location of the blow being raised to the burning temperature.

With a thick valve-rim, sufficient circumferential heat-flow capacity exists to carry the heat away from the leak aperture fast enough to prevent the temperature rising locally to the erosion or burning-point. The radial heat-flow capacity does not seem to be of as much importance as the circumferential heat-flow capacity. The Engineering Division has run tulip valves with thick rims and thin centers at full-throttle for many hours under conditions of continual local-blowing, without any burning and with only the slightest of scaling. Such tests have been carried out with valves of high-tungsten steel that burn readily if the design be poor.

That rim-burning is due to insufficient capacity to conduct the heat away from the point of leakage appears to be proved by the fact that a valve that is not rotating may have a large piece burnt out of the rim, and the cylinder seat at the location of the blow will not be scaled even slightly. This is due to the fact that the heat is abstracted from the cylinder seat so fast that the temperature cannot reach even the scaling-point.

In regard to items (3) and (4), the materials section of the Engineering Division has been able to determine by micrographic examination of its structure that, in a valve operating with a bright red-hot stem, a sharp change in the character of the structure occurs at approximately the point where the stem enters the guide,

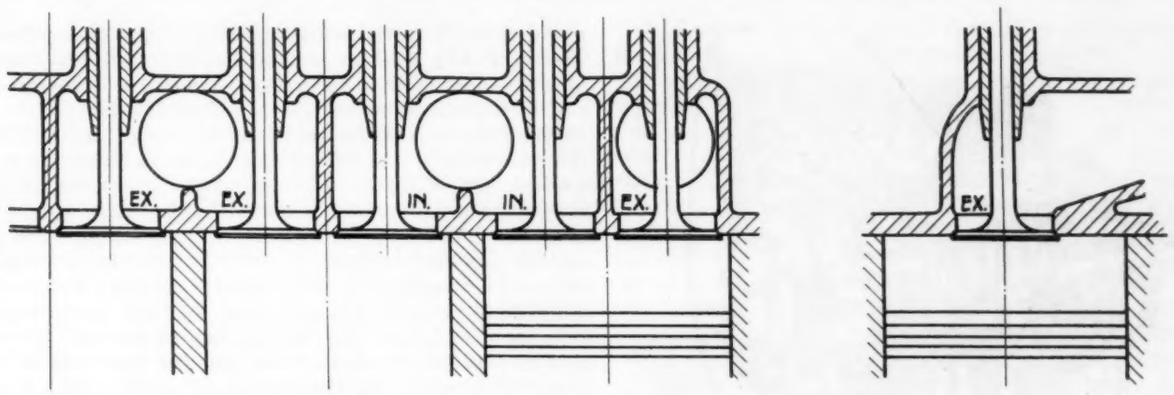

FIG. 11—PARTIAL SECTION OF AN ENGINE ILLUSTRATING POOR VALVE-COOLING CONDITIONS

the structure showing that the portion of the stem beyond the guide has reached a much higher temperature than the section shrouded by the guide. This evidence merely confirms the conclusions of observation and experience with aircraft engines. The subject is of sufficient interest to warrant reiterating the considerations of heat-flow from the valve-stem to the cooling medium, and of heat absorption within the exhaust-port by the valve-stem, guide and guide boss.

If the whole of the section of the valve-stem within the port be beyond the guide end, the whole of the axial flow of heat from the head has to be carried by the stem which is, in addition, absorbing heat from the exhaust gases throughout its exposed length. If the whole parallel section of the stem within the port be shrouded with the exposed guide, the conditions are somewhat better, provided the guide be of sufficient thickness to be able to conduct the heat axially without reaching a red heat. The exposed guide is better than the exposed stem, as the heat-flow capacity of both the stem and the guide is available for axial heat-flow to the port wall and the guide housing. The guide, of course, presents more surface to the flame than an exposed stem, but is more able to get rid of the heat absorbed, as this only has to pass a single thermal gap in flowing to the cylinder metal; whereas the heat from the stem has to pass two. The first is a bad gap, that occasioned by the loose fit of the stem in the guide; the second is the thermal gap caused by the fit of the guide in the guide boss, which is much less serious than the former because the guide is a drive or force fit in the guide boss.

The design in which the whole of the parallel length of the stem within the port is covered by the guide, the guide in turn being completely shrouded by a fairly thick guide-boss, has proved to be the most satisfactory in full-throttle service. Heat flowing along the stem begins to be dissipated rapidly to the guide at the point where the stem enters the guide. The guide, in turn, begins to dissipate the heat to the valve-guide boss. Thus the stem, the guide and the guide boss, are available for the axial transmission of the heat to the port wall. The heat-absorption effect of the large exposed surface of a long guide-boss having relatively a large diameter seems to be of little importance, as the boss is part of the port metal and the resistance to the heat-flow is small. Theoretically a large guide-boss appears to be bad on account of causing turbulence, thereby increasing the heat absorption; but, in practice, any ill effects due to this cause are completely nullified by the improved cooling obtained.

Valve-stems of small diameter, item (5), have a low axial heat-flow capacity and insufficent surface to transmit the heat to the valve-guide. Hence, they cause a greater temperature-gradient in the stem and a higher temperature-drop from the stem to the guide than exists with large-diameter stems.

The ideal cylinder valve-seat, item (6), is symmetrical, equally cooled at all points of its circumference and of such proportions that it will not distort due to circumferential inequalities of the temperature, if such occur, or uneven temperature-distribution in other portions of the head.

The design shown in Fig. 11 exhibits the undesirable characteristics, and is of a class found in many automobile engines. It is not surprising that this type is productive of distorted valve-seats, leading to the cracking and the burning of the seat and scaled and burnt valves. This design exhibits all of the objectionable features cited and, as regards the features producing seat and guide cooling, is not appreciably cheaper to produce than that shown in Fig. 12, which illustrates an exhaust-port section of the type-K cylinder. This valve layout, while admittedly involving extreme steps to secure the valve cooling, is nevertheless worth the price involved for aircraft purposes. The valve is internally cooled, resulting in the tip of the valve-stem running at much higher temperature than it would run otherwise. The hot tip of the valve is exposed to an oil-spray. Thus, a large amount of the heat is dissipated to the oil and the direct

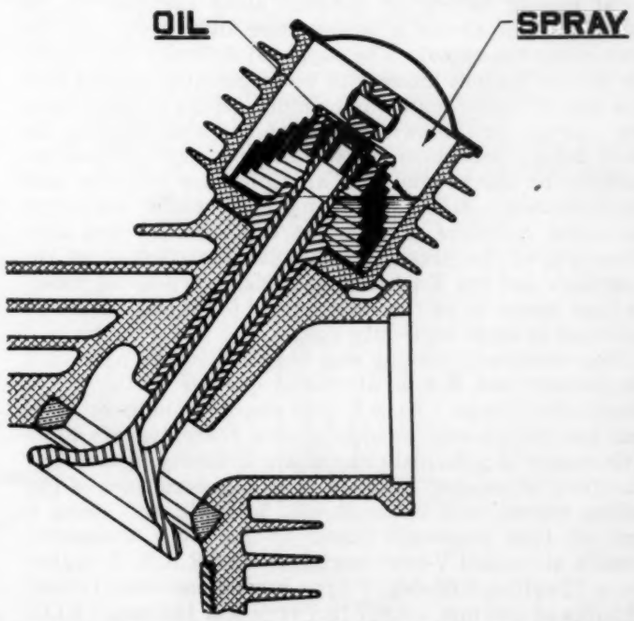

FIG. 12—SECTION OF THE TYPE-K EXHAUST-VALVE AND PORT

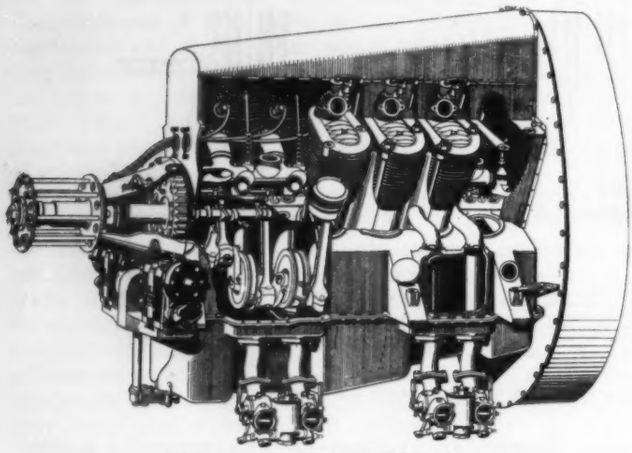

FIG. 13—THE R.A.E. 5 ENGINE

transmission of the heat from the stem to the guide is facilitated by the oil-film between the two. This design produces a remarkably efficient valve-cooling because, on continuous full-throttle in a cast-iron cylinder, the maximum valve temperature obtained is less than 1000 deg. fahr. and this is confined to a narrow band on the neck.

AIR SUPPLY

Few successful attempts have been made at cooling in-line engines with circumferential finning. Several engines with such finning have been produced in which no effort has been made to provide uniform air-distribution; a propeller-type fan is placed at the front of the engine and blows the air onto the front of the front cylinder and down the sides of the others. While such systems may function sufficiently well to allow the engine to pass muster in flat country, the result can scarcely be called cooling and disaster is likely to result from any attempt at protracted full-throttle running in hilly country. It is commonly considered that it is next to impossible to obtain anything like an even air-supply with an in-line engine having circumferential finning. The reason for this impression is by no means clear, for, if it is possible to secure a uniform air-supply with an in-line engine with axial finning having an air-duct along the tops of the cylinders, why should it be any more difficult with an air-duct along the sides? The supposed difficulty of air supply on car engines appears to be responsible for the general lack of individuality in cylinder design in this Country. As the best-known car-engine has axial finning, the stock-design procedure appears to be a copy of this, apparently on the ground that anything else is a leap into the unknown. Since all the most successful high-duty air-cooled cylinders produced in this Country and elsewhere are of the circumferential-fin type; that is, of the Lawrance and the Engineering-Division types, the axial-fin type seems to be the more risky, for it has yet to be produced in large high-duty sizes.

Very successful cooling was obtained in the past with the Renault and R.A.E. air-cooled engines of the 8 and 12-cylinder V-type. As a V-type engine is nothing more than two in-line engines side-by-side, there appears to be little reason to anticipate any grave difficulty in securing a uniform air-supply. Fig. 13 gives a general idea of the cooling system used in the R.A.E. 5 engine, this being a copy of that previously used on the very successful Renault air-cooled V-type engines. The R.A.E. 5 engine was a 12-cylinder 60-deg. V-type having cast-iron L-head cylinders of 100-mm. (3.937 in.) bore and 140-mm. (5.112 in.) stroke, with side inlet-valves in cages and superimposed exhaust-valves seated directly in the cylinder-head. The type was used in pusher airplanes and the cooling air was supplied by a centrifugal fan at the anti-propeller end of the crankshaft. The air from the fan volute discharged directly into a vertically tapering hood enclosing the top of the V and thence discharged between the cylinders at approximately 90 deg. to the crank axis. This air-supply arrangement, assisted by the blast due to the propeller inflow and the velocity of transition, gave reasonably efficient cooling in spite of what is now known to be an exceedingly crude cylinder design. The engine complete developed 140 net b.hp. and the fan absorbed 9.25 hp. or 6.50 per cent of the engine power. The excessive power-consumption of the fan was due to the crude cylinder design, the low output and the high resistance to air-flow over the cylinder-heads. For use in tractor airplanes, the fan was omitted and the hood reversed so that it tapered down to the anti-propeller end, the front plate being omitted and the rear end of the hood and the rear of the back cylinders being blanked-off. Thus, the air supply was scooped-up by the hood and discharged between the cylinders as in the pusher type.

The foregoing should suffice to show one tried method of cooling V-type engines, and it is now proposed to deal with the cooling-air supply of in-line engines, this being the type of most general interest to the car builder.

COOLING SYSTEM FOR IN-LINE ENGINES

Fig. 14 shows a design by the Engineering Division for a cooling system for in-line engines. This design is suitable for either aircraft or mechanical-transport vehicles, but is shown adapted to a passenger-car engine. It is merely an evolution of the Renault and R.A.E. methods applied to an in-line engine. The aim in this case has been to secure an approximately uniform air-distribution to all of the cylinders without the use of a large amount of cowling and baffles, both of these having decided disadvantages in destroying accessibility and, in the case of car engines, being potential rattle and squeak sources and, in aircraft engines, causing undesirable additional weight. The design shown uses the hood or engine covering as the collector and the ducts for the cooling air and divides this into two compartments along the vertical center line of the engine. This is carried out by a vertical partition arranged so that air-flow from one side of the engine to the other is possible only through the fin-cells.

Air is collected in front through a dummy radiator, the collection being probably assisted by a fan in the case of a car engine. The air is delivered from the collector duct to the sealed pressure-compartment, suitable baffles being arranged as shown to prevent the air piling up at the rear of the compartment and thus causing the major portion of the air to flow over the rear cylinders. The pressure compartment is sealed at the rear by the dashboard and at the bottom by a sill. The air flows from the pressure compartment to the exhaust air-compartment through the fin-spaces, all air being effectively used and none wasted, and thence away to the open under the floor-boards through holes in the sill or through suitable louvers in the hood. The design is arranged so that lifting the hood leaves the engine practically as accessible for adjustment and examination as in the normal water-cooled type.

BLAST DIRECTION

It has been shown previously that the method of application of the cooling blast to a circumferentially finned cylinder has a marked effect on the maximum temperature and the temperature distribution. It is manifestly

desirable to aim at holding the temperature at the hottest part of the cylinder to a minimum, and to secure such uniformity of the temperature distribution throughout the cylinder and the combustion-chamber walls as will eliminate distortion, axial temperature-differences are not productive of distortion but circumferential differences are. This result is suitably achieved in practice by directing the cooling blast against the exhaust side of the cylinder.

Tests made on a four-valve air-cooled cylinder, the results of which are given in Table 3 of the author's previous paper,[7] showed that the maximum combustion-chamber wall-temperature with inlet-side blast was 197 deg. fahr. higher than with the blast on the exhaust side and, further, that the circumferential temperature-difference at the top of the bore was increasel from 33 deg. fahr. with exhaust-side blast to 301 deg. fahr. with inlet-side blast. Uniformity of the circumferential temperature-distribution, and thus the minimum tendency for the cylinder bore to go out-of-round, is of considerable importance in securing silence of operation, not to mention other obvious advantages.

Exhaust-side blast, as it secures the maximum effect from a given air-supply by giving the major cooling-effect of the air to the portions of the cylinder having the maximum heat-reception, thus allows a minimum air-supply. Minimum air-supply is desirable both for economy of fan-power and as regards the rapid warming-up of the engine. With large aluminum aircraft-engine cylinders, satisfactory results are obtained with the blast directed at 90 deg. to the exhaust side. However, even in aircraft practice, with aluminum cylinders and the cooling efficiency not restricted by considerations of cost, if the maximum cylinder-size and minimum air-supply are aimed at, there is no question that exhaust-side blast is desirable.

Quantity of Air

As regards the quantity of air required for cooling, much will depend upon such circumstances as the average air-temperature in the locality where the car is to be used, whether the country is flat or mountainous, or if long periods of pulling through deep sand must be provided for. The quality of the cylinder design and the efficiency with which the air is used, that is, if the maximum effect of the cooling-air supply is given to the portions of the cylinder with the maximum heat-reception, will have an influence also upon the quantity of air necessary.

A. Ludlow Clayden states that, if a water-cooled car be provided with sufficient radiator-capacity to cool at full speed and full throttle, the cooling will be sufficient normally for all other circumstances. Whether such cooling-allowance will be suitable for all circumstances with an air-cooled car is not very certain. If an air-cooled car is to be operated continuously at full speed on the level, little difficulty will be experienced in cooling without a fan. Likewise, with cylinders of the type used on the B.S.A. car, no difficulty with over-heating will arise from continuous idling with a closed hood and no fan if suitable vents be provided in the hood to allow the escape of hot air, thereby allowing natural convection. With the B.S.A. engine it was possible to idle the engine for hours on end, without having the maximum cylinder-head temperature exceed 350 deg. fahr.

For continuous full-throttle work, approximately 30 cu. ft. of air per min. per b. hp. is sufficient with correct cylinder design. It is, however, unlikely that it will be necessary to provide a fan with such a capacity for the following reasons: (*a*) the fan will have its still-air capacity considerably augmented by the motion of the car; (*b*) the average load-factor of a car engine is well below 50 per cent and, for the normally limited periods of full throttle, the heat-storage capacity of the cylinders, crankcase, oil and the like is capable of providing a heat accumulator or flywheel which, for short periods, will store heat and prevent excessive cylinder-temperatures.

While the actual heat-storage capacity per degree of temperature-rise of an air-cooled engine is considerably lower than that of a water-cooled engine, the effective capacity is probably fully as great, due to the fact that the heat can be stored at a much higher temperature; while heat storage ceases in a water-cooled engine when boiling commences, no such critical temperature exists with an air-cooled type. To what extent the heat-storage capacity effect can be used is open to some doubt. The Engineering Division has had reason lately to modify its view in this respect, due to work on cast-iron cylinders.

Gibson has determined the characteristics of heat dissipation from finned cylinders in the wind-channel, and from actual engines found the relation between the speed of rotation and the air discharge from the fan and the cowling; he presents these data in his paper on The Air Cooling of Petrol Engines.[8] The results of these findings are as follows:

(1) The heat dissipation is proportional to the difference between the mean temperature of the cooling surfaces and that of the incoming air

(2) The heat dissipation is proportional to the 0.73 power of the weight of cooling air brought into

[7] See The Journal, April, 1922, p. 236.

[8] See *Proceedings of the Institution of Automobile Engineers*, vol. 14, p. 243.

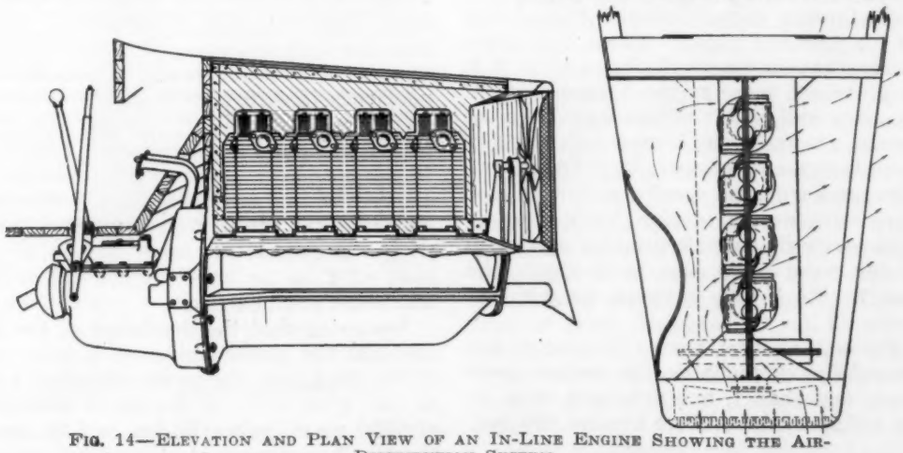

Fig. 14—Elevation and Plan View of an In-Line Engine Showing the Air-Distribution System

contact with the cooling surfaces per unit of time. For a pressure type of fan, this sensibly equals $V^{0.73}$, where V is the velocity of the cooling blast

(3) The discharge of a cooling fan of the propeller or centrifugal type, moving air through a cooling system of the type shown in Fig. 13, is approximately proportional to $N^{0.85}$, where N is the number of revolutions per unit of time

Thus, as the heat dissipated at a constant mean-temperature is proportional to $V^{0.73}$ and V varies as $N^{0.85}$, the heat dissipated will, at constant mean-temperature, vary as $(N^{0.85})^{0.73}$ or $N^{0.62}$.

Assuming that the mean effective pressure and heat loss per brake horsepower do not vary with the speed of rotation and, therefore, that the heat to be dissipated is proportional to the speed, or N, a fan-cooled engine will get hotter with an increase of the speed of rotation, since the heat to be dissipated will increase as N but the cooling effect at a constant mean-temperature will increase only as $N^{0.62}$. Under these conditions the following will hold:

Let

C_1 = a constant
C_2 = a constant
C_3 = a constant
N = the number of revolutions per unit time
T = the mean temperature-difference between the cooling surfaces and the incoming air
V = the velocity of the cooling blast

Then we have

$$\text{Heat given} = C_1N$$
$$\text{Heat dissipated} = C_2V^{0.73} \times T = CN^{0.62} \times T$$

Since the heat given equals the heat dissipated, we have

$$C_1N = C_2N^{0.62} \times T$$
$$T = C_1N/C_2N^{0.62} = C_3N^{0.38}$$

From the equation deduced with the stated conditions, the effect of doubling the speed of rotation of a fan-cooled engine at full throttle, when the fan capacity is not assisted by the car motion, will be to increase the mean temperature by 30 per cent. Actually, of course, the mean effective pressure falls off at the higher speeds; thus, the heat to be dissipated does not increase so fast as the engine speed. Further, the cylinder-wall temperature at the point of maximum temperature, even with a sensibly constant brake mean effective pressure and a constant air-supply, does not increase in direct proportion with the speed, or approximately with the output. This is of importance because, with sound cylinder-design, the maximum rather than the mean temperature appears to be the limiting factor in operation.

Table 1 shows the results of some tests made by the author on a four-valve air-cooled cylinder of 5½-in. bore and 6-in. stroke. Reference is made also to Figs. 1 and 9 and Table 10 of his previous paper.* These tests were carried out with a constant air-blast velocity of 87 m. p. h. The temperatures given are those at the hottest point of the combustion-chamber wall, and it will be seen, from the ratio of the temperature to the brake horsepower, that the temperature does not increase proportionally to the output. The temperature increase with the speed was in this case greater than occurs with normal design, owing to the restricted exhaust capacity that resulted in the maximum heat-reception by the point least able to dissipate the heat received; that is, the bridge between the two exhaust-valves.

The figures in Table 1 suggest that a fan-cooled engine with an air-supply proportional to the engine speed to a fractional power less than 1, and arranged so as to keep the maximum cylinder-temperature around 400 deg. fahr. with full throttle at normal engine-speed, will not develop excessive cylinder-temperatures at full speed and full throttle. This is in view of the Engineering Division's experience that temperatures hitherto considered excessive can be used without damage, reduction of output or unsatisfactory performance.

TABLE 1—ENGINE TESTS

Speed, R.P.M.	Power, B.Hp.	Brake Mean Effective Pressure, Lb. per Sq. In.	Fuel-Consumption, Lb. per B.Hp-Hr.	Difference between Cylinder-Head Temperature and Air Temperature, Deg. Fahr.	Ratio, Temperature to Brake-Horsepower	Per Cent
1,200	25.4	117.5	0.61	389	15.3	100
1,400	31.0	123.0	0.60	407	13.1	86
1,600	36.9	128.0	0.57	450	12.2	80
1,700	39.5	129.0	0.56	452	11.4	75
1,800	40.9	126.0	0.56	462	11.3	74
1,900	42.1	123.0	0.56	463	11.0	72
2,000	43.1	119.5	0.56	515	12.0	79

To arrive at the quantity of air required for cooling, it is proposed to use a hypothetical design of a six-cylinder car-engine, using a known cylinder arranged on a crankcase in a manner that past experience indicates fairly definitely will produce satisfactory cooling with the air-supply and output assumed. The B.S.A. cylinder has been taken as a basis, the design being modified to suit an in-line engine, the fins being slabbed off to be 11/16 in. long on the sides at the base of the barrel, a 1/16-in. space between the fin-tips being allowed. Two cylinders are shown arranged thus in Fig. 15 and, if all the air used be forced through the fin-cells, the area available for flow is 8.6 sq. in. The bore is 3½ in. and the stroke 3⅜ in., giving a capacity for six cylinders of 195 cu. in. The output is assumed to be 100-lb. per sq. in. brake mean effective pressure at 2500 r.p.m., or 61.5 b. hp., and the blast velocity 60 m.p.h., or 5280 ft. per min., at the throat as shown by the black area in Fig. 15.

The total air-flow under these conditions becomes $[(8.6 \times 6)/144] \times 5280 = 1890$ cu. ft. per min.; and this figure equals 30.8 cu. ft. per min. per b. hp. The theoretical power necessary to move this quantity of air at 60 m.p.h. is 0.52 hp. It will, however, be of more interest to calculate the power required from the pressures found necessary on engines that have been poorly laid-out.

The R.A.E. 5 engine shown in Fig. 13, which had an excessive pressure-drop in the air-flow around the cylinders, required a total head of 6 in. of water. The static plus the velocity head, in the cowl to produce approximately the 60-m.p.h. blast between the cylinders that was necessary to secure good cooling at continuous full-throttle on the dynamometer. A total head of 6 in. of water will therefore be assumed, although this is probably at least 50 per cent in excess of that necessary. By extrapolation of the tables given in Kent's Mechanical Engineers' Pocket-Book for the American Blower Co.'s Sirocco fans, the power required to force 1890 cu. ft. of air with a total head of 6 in. or water is 3.1 hp. or 5 per cent of the maximum engine-output.

Assuming that the discharge of the fan varies as $N^{0.85}$ and that the power consumed is proportional to the cube of the discharge, the power absorbed will be proportional to $(N^{0.85})^3$ or $N^{2.55}$. If 3.1 hp. is absorbed at 2500 r.p.m., at 1250 r.p.m. only 0.53 hp. or 1.70 per cent of the full-throttle horsepower at this speed, assuming 100 lb. per

*See THE JOURNAL, April, 1922, pp. 233, 237 and 257.

sq. in. brake mean effective pressure, will be absorbed. But 1.7 per cent of the full-throttle output is equivalent to only 1.7 lb. per sq. in. friction mean effective pressure, a power loss that is negligible in comparison with the other losses, especially when it is considered that the fan and water-pump losses of the normal water-cooled engine are replaced by the above loss. It is easily possible to produce an extrafriction-loss of three or more times the order of 1.7 lb. per sq. in. mean effective pressure by having the oil too cold or too heavy.

Objection may be raised to the above-given computations as being hypothetical. However, as good cooling with similar blast-velocities and cylinder-spacing has been obtained with aircraft engines and, as the output assumed has actually been obtained with this cylinder, although without the fins slabbed-off but with a lower cooling-blast velocity, the whole appears not to be wild assumption.

The B.S.A. cylinder is not of the most useful proportions for use on an in-line engine, as the bore-stroke ratio used will produce a relatively long engine. However, no difficulty would arise in using a cylinder of about 3-in. bore and 5-in. stroke in a similar manner. The hypothetical design considered would be capable of continuous full-throttle at the speed quoted and, since the air was considered as being supplied without assistance from the motion of the car, there is no question that the engine would be grossly over-cooled in normal passenger-car service.

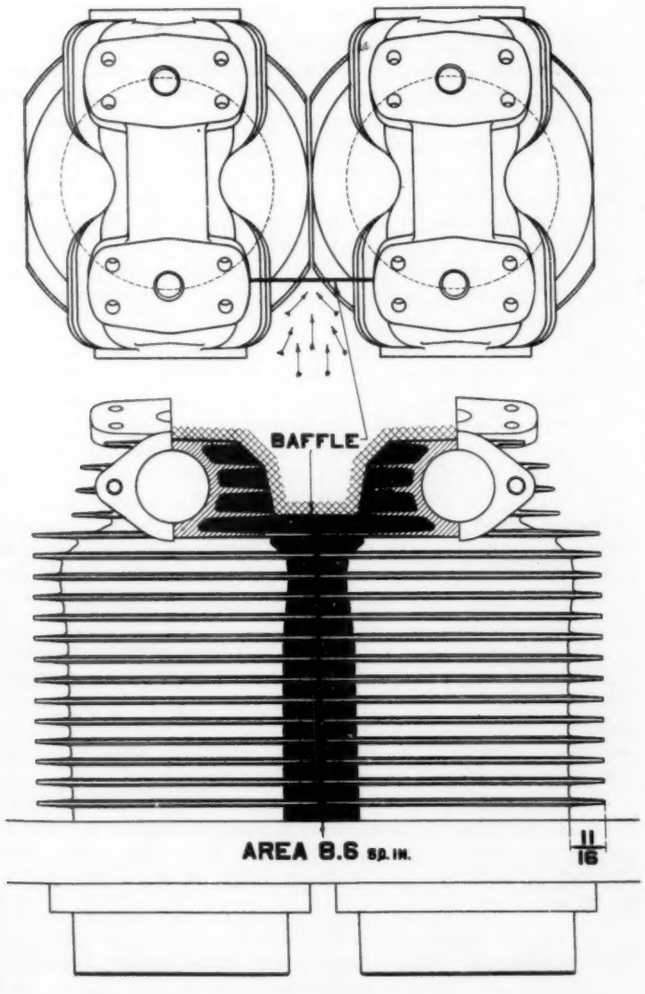

FIG. 15—DRAWING SHOWING THE AREA AVAILABLE FOR AIR-FLOW BETWEEN IN-LINE ENGINE CYLINDERS

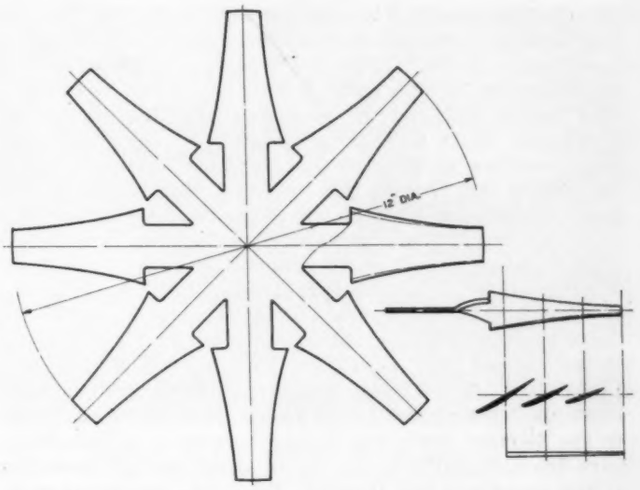

FIG. 16—DRAWING OF BLOWER DEVELOPED BY THE ENGINEERING DIVISION

FANS

The pressure type of fan is decidedly more efficient for cooling than the suction type, for the turbulent discharge of the pressure type is much more effective for heat abstraction than the approximately streamline flow induced by the suction type. Further, with the suction type at a given fan-speed, the hotter the engine becomes, the lower the weight of air passing over the cylinders will be, the discharge of this type falling much more rapidly under such conditions than that of the pressure type.

The use of an exhaust turbine-driven cooling-fan has been proposed. While this would doubtless be very expensive and difficult to produce, it would be nearly ideal in that the air-supply would be almost directly proportional to the indicated horsepower developed at any moment and would further provide a very efficient muffler, although the fan itself would be liable to scream on account of the high rotational speed necessary.

As regards the choice of the fan type, whether propeller or centrifugal, much will depend upon the service required. For a truck or tractor engine that must be capable of developing continuous full-power at very low vehicle-speeds, there is little question that the centrifugal type is the best, since it handles large volumes of air with a relatively high discharge-pressure at a fairly low number of revolutions per minute and is relatively very quiet.

For passenger-car use, there is much to be said in favor of the propeller type of fan because it can be used in the conventional position, is easy to drive, has, when properly designed, a very high efficiency, does not require any elaborate casing to secure efficiency and, when suitably installed in a phaeton, will have its still-air capacity very considerably augmented by the motion of the car. The propeller type is, however, noisier than the centrifugal type and, when running at very high speeds, is liable to produce an objectionable whine. The propeller type, unless of large diameter or run at very high speed, does not give as high a discharge pressure as the centrifugal type, and is therefore not so suitable where cooling is required at continuous full-power with a slowly moving vehicle.

If a propeller type of fan be used, care will be neces-

sary in the design of the cooling system to keep the friction-head of the cowling, the fin-cells and the ducts to a minimum. There is little question that a propeller type of fan would be of small use with an axial fin-construction having long narrow cooling-cells with a large friction-head. With a centrifugal fan, a high friction-head in the cooling system can be overcome by an increased fan-power, bad though this may be. With the propeller type, however, a high friction-head will result in little or no air-flow. Some propeller-type blowers recently designed and produced by the propeller branch of the Engineering Division are of considerable interest from the air-cooled passenger-car engine point of view. One of these blowers is illustrated in Fig. 16. This blower runs at 4000 r.p.m., has a delivery of 2000 cu. ft. per min. against 0.85 in. of water pressure, absorbs 0.36 hp. and shows an efficiency of 74 per cent. This particular point on the delivery curve was chosen because it is considered to be the most suitable one for an air-cooled car operating at full speed and full throttle. While the discharge pressure of this fan is not very high, it must be considered that, at high speeds, the head will be considerably augmented by the velocity of transition of the car. A blower of the type shown is in a very different class from that of the wind-churns fitted to the average water-cooled car. The average fan fitted is extremely inefficient and not appreciably cheaper than the one shown in Fig. 16. Fans absorbing 6 hp. at the maximum engine-speed are by no means unknown on water-cooled cars. A power consumption of this order constitutes a serious wastage without adequate return.

Fuel

Probably owing to the fact that their mean wall-temperatures are normally somewhat higher, an air-cooled cylinder seems to require fuel with less tendency to detonate than does a water-cooled engine of similar size and compression-ratio. The difference is not very marked and, in any case, it is probable that detonation is apparently more acute in the air-cooled type than in a similar water-cooled cylinder, due to the fact that the noise is not deadened by the equivalent of a water-jacket. However, the air-cooled engine is not damaged by detonation to anything like the extent that the water-cooled type is. The Engineering Division has run air-cooled cylinders for extended periods of full throttle with violent ringing pinking occurring every second or third explosion without apparent damage to the cylinder and with only slight distress from the spark-plugs. Such treatment of a water-cooled cylinder will result usually in exhaust-valve, spark-plug and piston trouble.

Spark-Plugs

The Engineering Division has had no great difficulty in securing spark-plugs capable of standing-up to sustained full-throttle in air-cooled engines. Plugs of both mica and porcelain construction have given satisfaction, and there seems to be no ground for the notion that highly elaborate and expensive plugs are essential or that over-heated and burnt-out plugs are a necessary adjunct of an air-cooled engine. Plugs capable of standing more than 100 hr. of full-throttle running without apparent deterioration have been obtained by the Division. These plugs are not of a type designed only to withstand heat, but have considerable resistance to fouling in addition to being able to withstand unlimited full-throttle running.

Pistons

The use of piston material other than aluminum, and possibly magnesium alloys in the future, for air-cooled car engines is not worthy of consideration, as the effect of the aluminum piston, in comparison with cast iron or steel, is to secure an increase of output, a lower fuel-consumption, a higher allowable compression, less tendency to detonation and a better pick-up, particularly at low speeds. Any company that is incapable of successfully using aluminum pistons would be well advised to leave air-cooled cars alone. There is no reason to anticipate piston trouble with air-cooled engines. The Engineering Division has had much less burnt piston trouble with air-cooled than with water-cooled engines. In fact, anything other than the very slightest of pitting has yet to be experienced, in spite of several hundreds of hours of full-throttle testing with cylinders of 5½-in. and larger bore.

Collapsed and severely pitted pistons are by no means unknown with the modern high-efficiency water-cooled aircraft-engine, and just why this should not occur also with air-cooled engines of similar duty and output is not clear; nevertheless, such is the case. It is possible that the lack of piston trouble in the air-cooled aircraft-engines is due to the more careful general thermal design of the whole cylinder-unit that has had to be developed to make a cylinder of high output function at all.

Conclusion

While, from the automobile standpoint, the paper may seem to deal too largely with large-capacity cylinders and full-throttle conditions, nevertheless the following reasons justify such considerations. First, scaling-down of design in internal-combustion engine cylinders is a safe process, while scaling-up is decidedly unsound. The dangers of scaling-up would be speedily evident if an attempt were made to use the type of cylinder design generally found in air-cooled vehicles, for 150-cu. in. aircraft-engine cylinders. The use of scaled-down successful aircraft practice has, in general, proved mechanically and thermally satisfactory. Much of the so-called aircraft practice grafted onto post-war car-design has been the work of designers lacking in experience with successful modern aircraft engines. A considerable amount of the practice can rightly be described as aircraft design, with the qualifying statement that it was proved unsound for aircraft use and was discarded during 1914 and 1915. Second, as thermal troubles have been the greatest cause of difficulty with air-cooled cars, a consideration of the designs capable of continuous full-throttle running is likely to demonstrate the ease with which the relatively mild thermal difficulties of an air-cooled car can be overcome.

The preparation of this paper has been greatly assisted by the able work of H. E. Morehouse in preparing many of the designs illustrated and by the very practical automobile viewpoint of L. H. Pomeroy who has enabled the author to grasp more clearly the essential differences in the duty of automobile and aircraft engines.

The appendices by Messrs. Dix and Taylor should tend to give the contribution a less biased character, for these gentlemen are thoroughly familiar with the vices and virtues of both air and water-cooling, in their own fields of effort.

APPENDIX 1

Experimental Development of Air-Cooled Engine-Cylinders[10]

In connection with Mr. Heron's paper, it is of interest to review briefly the methods used by the Engineering

[10] Contributed by C. Fayette Taylor, S.M.S.A.E.—Engineer in charge of powerplant laboratory, engineering division, Air Service, McCook Field, Dayton, Ohio.

Division in developing and testing cylinder units for both air and water-cooled engines, and to discuss some of the results obtained with several of the air-cooled cylinders mentioned by Mr. Heron. In general, nearly all the conclusions reached in Mr. Heron's paper regarding the performance of air-cooled cylinders have been confirmed by work in the Engineering-Division laboratories. In those few cases where tests have not been made by the Engineering Division, the information given is based on tests made by the Royal Aircraft Establishment of Great Britain.

All cylinder units developed by the Engineering Division are tested on an apparatus known as a universal-test engine. This consists of a ruggedly built single-cylinder crankcase arranged so as to accommodate any of the usual sizes of aircraft-engine cylinder with any valve-gear, of either the push-rod or the overhead type. The crankcase is equipped with a chankshaft having an adjustable stroke, and with means whereby the compression-ratio of the cylinder can be changed readily. The lubrication system of the engine is arranged so that the oil is circulated through temperature-control apparatus, by which any desired temperature of the lubricating oil can be maintained. Means are provided also for adjusting the rate of flow of the lubricating oil through the bearings and the crankpin of the engine and for regulating, within limits, the amount of oil sprayed into the cylinder.

The universal-test engines, several of which are available at the Engineering Division laboratories, are direct-connected to 50-hp. Sprague electric dynamometers of the usual type. Fig. 17 shows an installation especially designed for testing air-cooled cylinders. The cooling-air is supplied by a 100-hp. centrifugal-blower delivering air through a duct 18 in. square at velocities up to 120 m.p.h. The blower is arranged so that the blast can be applied against the cylinder in a direction either parallel or perpendicular to the crankshaft axis. The fuel-consumption is measured by accurately graduated special volume-meters. Stop-watches are used to time the flow of fuel through the meters. Load scales, speed-counters, thermometers, air-speed meters and the like, are installed carefully and calibrated at regular intervals. Cylinder temperatures are obtained from thermocouples that are embedded in the cylinder-walls.

In testing an experimental cylinder-design, at least two cylinders are built, since it has been found that results obtained on only one cylinder of a type are apt to require verification. Great care is exercised in "working in" cylinders for test, to make sure that the pistons, piston-rings, valves and valve gear are thoroughly worn-in and that the cylinder castings are well seasoned. This eliminates a frequent source of error in testing new cylinders. Considerable importance is attached also to the determination of the proper carbureter and venturi size for a given cylinder. While it is possible to estimate these sizes closely, a series of runs with different sizes is made, where any doubt exists, to establish the proper conditions in the induction system.

PERFORMANCE INVESTIGATIONS

In making a thorough investigation of the performance of an experimental cylinder, the necessity of changing the mixture-ratio and the timing of the ignition, to suit varying conditions, is realized fully. When investigating the performance of a cylinder at any given speed, the first run usually made is one in which the amount of fuel supplied to the cylinder is varied and all other conditions are held constant, except that the timing of the spark must be set to the optimum position for the particular mixture-ratio used. Curves of typical runs of this kind are shown and will be discussed in greater detail later. Such curves are extremely convenient in determining the proper carbureter and ignition settings for a certain cylinder at any given speed. In the case of air-cooled cylinders where the performance of the cylinder is so closely connected with the amount of fuel supplied, tests of this kind are of primary importance.

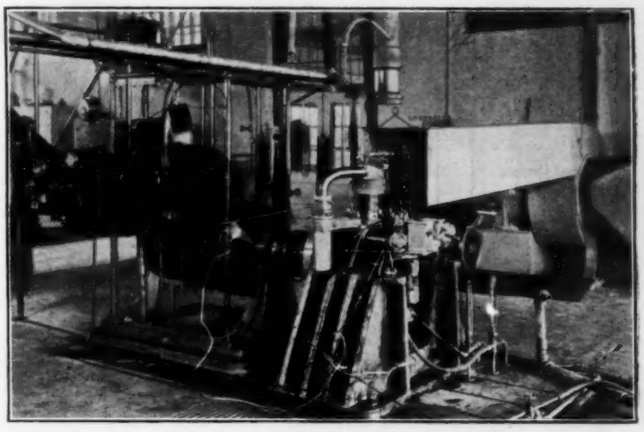

FIG. 17—INSTALLATION OF THE SINGLE-CYLINDER TEST APPARATUS USED BY THE ENGINEERING DIVISION

One particularly difficult problem in single-cylinder development is that of making proper allowance for the friction losses on the single-cylinder engine. To predict accurately the performance of a multi-cylinder engine from the single-cylinder tests, it is necessary to determine as closely as possible the friction losses of the single-cylinder engine. Unfortunately, no direct method of measuring friction losses during actual engine operation exists at present. The following methods for obtaining an approximation of the friction losses are available:

(1) The engine can be driven by the dynamometer at various speeds, holding the oil temperature, as well as the water temperature in the case of a water-cooled cylinder, at the same value that is used in the running tests and observing the power required to drive the engine at various speeds. This method is open to the objection that the piston and the bearing pressures are not equal to those that exist during the firing stroke, and that the cylinder is not running at its normal working-temperature. The difference in the bearing pressures would tend to make the readings thus obtained too low, while the lower cylinder-temperatures would tend to make them higher than during normal operation. It is probable that this latter effect is the more important, and that the friction losses so obtained are slightly higher than those actually existing when the engine is running

(2) The engine can be run under normal conditions at a given speed until the conditions are steady, the ignition and air-blast then being suddenly switched-off and the dynamometer operated as a motor. As soon as possible after this shift is made, a number of readings of the friction load are taken at regular intervals of time, measured from the instant at which the engine ceased to fire. The results are plotted in terms of friction load versus time, which gives a curve that can be extrapolated to the point of zero time, the intersection being the desired friction-load. This method attempts to take account of the normal operating-temperatures

TABLE 2—FRICTION LOSSES; DETERMINED ON THE UNIVERSAL-TEST ENGINE[11]

Cylinder-Type Designation	Speed, r.p.m.	Universal[12]	*K*	*W*-1	*H*	*W*-2	*L*	Cast-Iron Head	*J*	*S*
Bore, in.			4½	5½	5½	6½	5	5	5⅝	5½
Stroke, in.			5½	6½	6½	7½	7	6	6½	6½
Type of Cooling			Air	Water	Air	Water	Water	Water	Air	Air
Compression-Ratio			5.32	5.40	5.40	5.43	5.42	5.38	5.30	5.21
Total Friction, hp.	1,600	2.75	5.20	9.00	6.40	13.10	8.30	7.20	9.00	8.00
	1,700	3.00	5.82	10.00	7.40	15.30	9.00	7.85	10.00	8.90
	1,800	3.20	6.48	11.00	8.50	18.00	9.80	8.50	.....	9.90
Friction Mean Effective Pressure, lb. per sq. in.	1,600	.....	29.40	28.70	20.40	26.00	29.80	30.20	27.50	25.50
	1,700	.....	31.00	30.00	22.20	28.60	30.40	31.00	28.70	26.70
	1,800	.....	32.60	31.20	24.20	31.80	31.30	31.70	.....	28.10
Total Friction without Crankshaft and Auxiliaries, hp.	1,600	.....	2.45	6.25	3.65	10.35	5.55	4.45	6.25	5.25
	1,700	.....	2.82	7.00	4.40	12.30	6.00	4.85	7.00	5.90
	1,800	.....	3.28	7.80	5.30	14.80	6.60	5.30	.....	6.70
Friction Mean Effective Pressure without Losses of Crankshaft and Auxiliaries, lb. per sq. in.	1,600	.....	13.85	19.92	11.64	20.60	19.95	18.65	19.10	16.75
	1,700	.....	15.00	21.00	13.20	23.10	20.30	19.25	20.10	17.70
	1,800	.....	16.48	22.10	15.00	26.20	21.10	19.75	.....	19.00

[11] Different cylinders used, of types designated.

[12] Inclusive of crankshaft and auxiliaries.

of the engine but, in case of the universal-test engines, it gives results very nearly the same as are obtained by method (1)

(3) The indicated horsepower can be obtained by a suitable indicator and the friction losses computed by subtraction of the brake horsepower. This is obviously the most desirable method but, unfortunately, the Engineering Division has not yet discovered an indicator that is sufficiently accurate for this purpose at the high speeds at which aviation engines are run. It is to be hoped that such an instrument will be developed in the near future

Method (1) is generally used by the Engineering Division, since it is very convenient and appears to be as accurate as any of the methods available. Several independent investigators have stated that the results obtained by this method are very close approximations of the friction losses that exist under actual running conditions.

Table 2 shows the friction losses of the universal-test-engine crankshaft and auxiliaries, which are practically the same irrespective of the type or the size of the cylinder. Table 2 also shows the friction losses of a number of different types of cylinder on the engine, both including and excluding these losses. For comparing the performance of different types and sizes of cylinder, the writer prefers to add the power lost in friction of the basic engine, that of the crankshaft and the auxiliaries, to the brake power-output of the engine with each cylinder, and to correct the performance data to the values so obtained. While this method gives better performance-figures than can be obtained normally in multi-cylinder engines, it is useful for comparative purposes, and the absolute values so obtained are a measure of a cylinder's potential output since only those losses inherent in the cylinder design are included; those losses that depend solely on the particular type of crankcase on which the cylinder happens to be mounted are excluded. Thus, the data obtained by this method show the performance that can be *approached* with a multi-cylinder design by reducing the induction-system losses, and the friction losses in the crankshaft and the auxiliaries, to a minimum. This method of correcting for friction losses was used in preparing some of the illustrations, and gives results that are slightly less favorable to the cylinder than would be obtained by the method used in the paper. The average distribution of the so-called "friction" losses in the modern V-type water-cooled aircraft engine is as shown in Table 3.

TABLE 3—DISTRIBUTION OF FRICTION LOSSES

Distribution	Indicated Power, Per Cent	Friction Losses, Per Cent
Pumping Losses	5.85	45.0
Piston, Piston-Ring and Connecting-Rod Friction	6.24	48.0
Crankshaft and Auxiliaries, including Valve-Gear	0.91	7.0
Total Friction Losses	13.00	100.0

MULTI-CYLINDER PERFORMANCE

In estimating multi-cylinder performances from single-cylinder test-results, factors other than friction losses must be taken into account. The single intake-pipe of the single-cylinder installation, together with the fact that lower induction-system depressions can be used due to the absence of the considerations of distribution, idling and acceleration, results in an indicated power-output for the cylinder on the universal-test engine considerably greater than that of the same cylinder on a multi-cylinder engine. With large cylinders, the lower induction-system losses often more than offset the high friction-losses of the single-cylinder crankcase, and result in brake power-outputs from the single-cylinder installations that exceed the brake power-outputs of the same cylinders on multi-cylinder engines.

Fig. 18 illustrates the difference between single and multi-cylinder performance for a 5 x 7-in. cylinder. It will be noted that the single cylinder gives a higher brake power-output, a much higher indicated-output and a higher "peak" on the power curve, than the same cylinder on a multi-cylinder engine. In the case of a smaller cylinder, such as the type-K shown in Fig. 1, it is probable that the uncorrected brake power-output obtained on the universal-test engine would be exceeded in a well-designed multi-cylinder installation. However, so many variable factors are involved, such as the number and the arrangement of the cylinders and the different valve-timing and manifolding required on the multi-cylinder engine, that no accurate method of predicting multi-cylinder performance from a single-cylinder test has yet been evolved. With certainty, it can be said only that the results obtained with the methods of correction used in Appendix 1 or in the paper have been approached, but not yet equaled, in the best multi-cylinder designs.

OTHER INVESTIGATIONS

In addition to determinations of the effect of the mixture-ratio on the performance and of the magnitude of the friction losses, an investigation of the power-output at various rotational speeds is conducted usually on an experimental cylinder. Also, in the case of an air-cooled cylinder, an investigation is made of the performance with varying air-blast velocities and with varying air-blast direction. For a water-cooled cylinder, an investigation is made also of the performance with varying cooling-water temperatures and rates of flow, and one for endurance qualities for long periods of full-throttle operation.

Any or all of the above investigations can be conducted with different compression-ratios and valve-timing. All of the single-cylinder work is done at full throttle, since at throttled speeds the wide departure in the induction system from multi-cylinder conditions makes the results of little value.

Fig. 19 shows comparative performance curves of a typical water-cooled and an air-cooled cylinder, at con-

stant speed and with varying mixture-ratios. The importance of this particular type of curve cannot be overestimated in single-cylinder work since, with a given compression-ratio and speed, it indicates clearly the points of maximum efficiency and maximum power-output of a given cylinder. It will be noted that the compression-ratio of the two cylinders is approximately the same and that, therefore, the results can be compared safely. The water-cooled cylinder evidently is capable of satisfactory operation on leaner mixtures and with lower specific fuel-consumption than the air-cooled cylinder. However, as the mixture becomes richer, the air-cooled cylinder compares

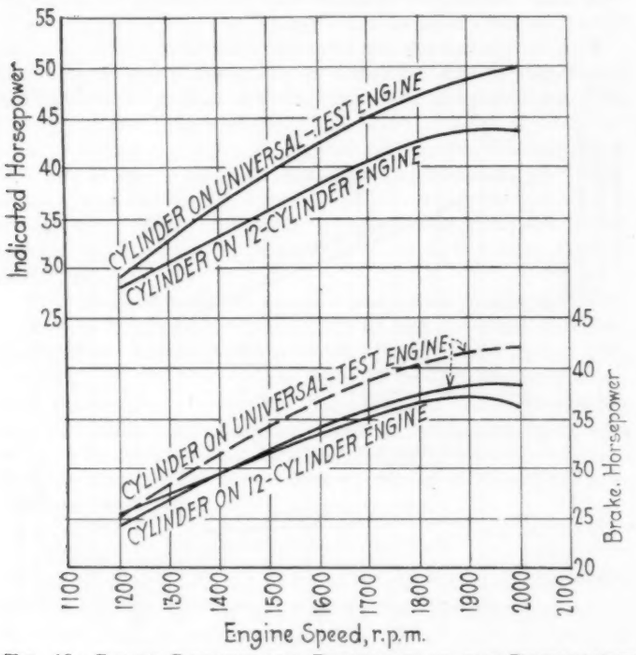

FIG. 18—CURVES SHOWING THE DIFFERENCE IN THE PERFORMANCE OF A SINGLE-CYLINDER AND A MULTI-CYLINDER ENGINE USING THE SAME CYLINDER DESIGN

more favorably with the water-cooled type. It is interesting to note that, with the leanest mixture at which the air-cooled cylinder will run, the difference between the temperatures of the two cylinders, measured at the hottest point, was only 200 deg. fahr. and, with the mixture-ratio decreased to 11.3, this difference was reduced to approximately 130 deg. fahr. In several instances it has been found possible to reduce the fuel consumption of a similar air-cooled cylinder to as low as 0.453 lb. per b. hp-hr., but this could be done only under favorable conditions and with a considerable reduction of power-output. The curves of Fig. 19 are, therefore, considered typical. The difference in the mean effective pressure developed by the two cylinders is due to differences in their design, and is not attributable to the different method of cooling. A higher mean effective pressure for the water-cooled cylinder would result in only slightly higher temperatures.

Fig. 20 is an interesting comparison between two air-cooled cylinders of identical size and method of construction; one has a hemispherical head with two valves, and the other a flat head with one intake and two exhaust-valves. These are respectively the type-J and type-I cylinders illustrated in Mr. Herron's previous paper.[13] It will be noted that the hemispherical-head cylinder shows a lower fuel-consumption, a higher power-output and a lower

[13] See THE JOURNAL, April, 1922, p. 231.

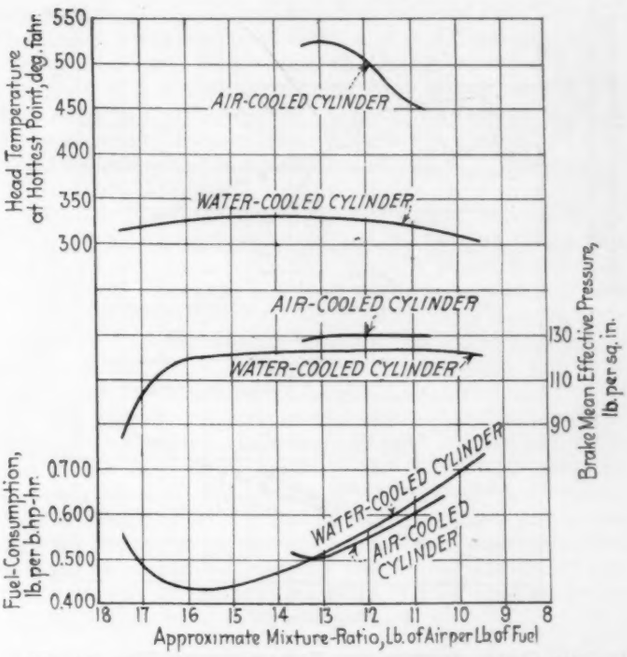
Head Temperature at Hottest Point, deg. fahr.
Brake Mean Effective Pressure, lb. per sq. in.
Fuel-Consumption, lb. per b.hp-hr.
AIR-COOLED CYLINDER
WATER-COOLED CYLINDER
AIR-COOLED CYLINDER
WATER-COOLED CYLINDER
WATER-COOLED CYLINDER
AIR-COOLED CYLINDER
Approximate Mixture-Ratio, Lb. of Air per Lb. of Fuel

FIG. 19—COMPARATIVE PERFORMANCE CURVES OF A 5 X 7-IN. WATER-COOLED CYLINDER AND A 5½ X 6½-IN. AIR-COOLED CYLINDER

Type of Cylinder	Air-Cooled	Water-Cooled
Compression-Ratio	5.3	5.4
Water-Outlet Temperature, deg. fahr.	...	170
Air Temperature, deg. fahr.	32	90
Air-Blast Speed, m.p.h.	100	...
Speed of Engine, r.p.m.	1,650	1,700

head-temperature than the flat-head cylinder tested under the same conditions. This bears out very strikingly the statements made in the paper in regard to the shape of

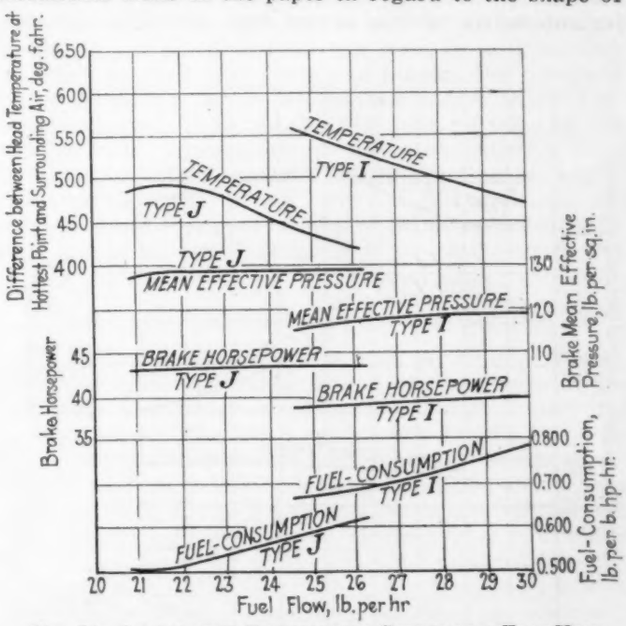
Difference between Head Temperature at Hottest Point and Surrounding Air, deg. fahr.
Brake Horsepower
Brake Mean Effective Pressure, lb. per sq. in.
Fuel-Consumption, lb. per b.hp-hr.
TEMPERATURE TYPE I
TEMPERATURE TYPE J
TYPE J MEAN EFFECTIVE PRESSURE
MEAN EFFECTIVE PRESSURE TYPE I
BRAKE HORSEPOWER TYPE J
BRAKE HORSEPOWER TYPE I
FUEL-CONSUMPTION TYPE I
FUEL-CONSUMPTION TYPE J
Fuel Flow, lb. per hr

FIG. 20—COMPARATIVE PERFORMANCE CURVES OF A FLAT-HEAD THREE-VALVE AIR-COOLED CYLINDER AND A HEMISPHERICAL-HEAD TWO-VALVE AIR-COOLED CYLINDER

Type of Cylinders	I	K
Form of Cylinder-Head	Flat	Hemispherical
Compression-Ratio	5.30	5.37
Diameter of Intake-Valve, in.	2⅛	2½
Lift of Intake-Valve, in.	⅜	⅜
Number of Exhaust-Valves	2	1
Diameter of Exhaust-Valves, in.	1½	2¼
Lift of Exhaust-Valves, in.	⅜	⅜
Speed of Engine, r.p.m.	1,650	1,650
B'ast Speed, m.p.h.	100	100
Air Temperature, deg. fahr.	74	32

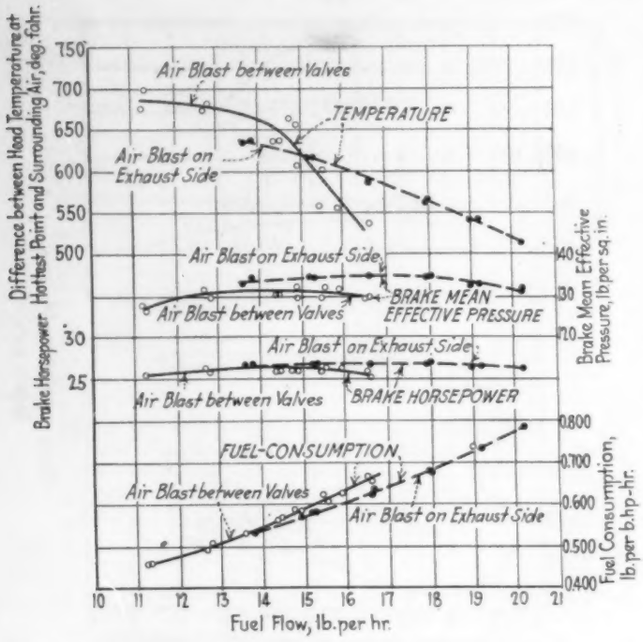
Air Blast between Valves
TEMPERATURE
Air Blast on Exhaust Side
Air Blast on Exhaust Side
BRAKE MEAN EFFECTIVE PRESSURE
Air Blast between Valves
Air Blast on Exhaust Side
Air Blast between Valves
BRAKE HORSEPOWER
FUEL-CONSUMPTION
Air Blast between Valves
Air Blast on Exhaust Side
Difference between Head Temperature at Hottest Point and Surrounding Air, deg. Fahr.
Brake Horsepower
Brake Mean Effective Pressure, lb. per sq. in.
Fuel Consumption, lb. per b.hp-hr.
Fuel Flow, lb. per hr.

FIG. 21—CURVES SHOWING THE VARIATION WITH FUEL FLOW AND WITH TWO DIFFERENT BLAST DIRECTIONS IN THE PERFORMANCE OF A TYPE-K CYLINDER

the combustion-chamber. These results are uncorrected for the friction losses, since both cylinders were tested on the same universal-test engine and dynamometer. The left-hand end of each curve in Fig. 20 represents the point beyond which the fuel-flow could not be reduced without causing a severe preignition and immediate power-drop. The most interesting material for cylinder construction for automotive vehicles is cast iron, due to its low cost, ease of casting and the absence of the inserted valve-seats and spark-plug bosses required with aluminum.

Figs. 21, 22 and 23 should be of especial interest to the automobile engineer, since they represent actual test-results on a cylinder with cast-iron head of the type illustrated in Fig. 1 of the paper. While this cylinder has a steel barrel, it could be made easily with an integral cast-iron barrel having cast-iron fins without affecting the performance materially. Tests made by the Engineering Division indicate that the material of the barrel below the top of the piston-travel does not have any great effect on the cylinder performance, provided it is reasonably well finned and there is no serious obstruction to the heat-flow from the head to the barrel.

Fig. 21 illustrates the performance of the type-K cast-iron-head cylinder having a compression-ratio of 5.32 to 1, with varying rates of fuel-flow and with two differ-

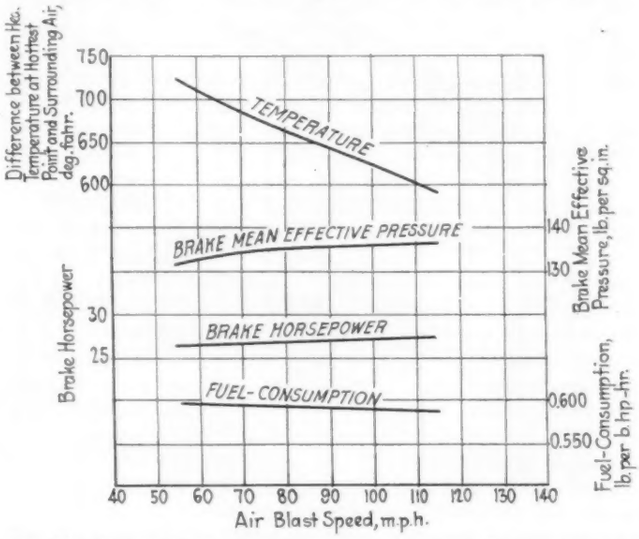
TEMPERATURE
BRAKE MEAN EFFECTIVE PRESSURE
BRAKE HORSEPOWER
FUEL-CONSUMPTION
Difference between Head Temperature at Hottest Point and Surrounding Air, deg. Fahr.
Brake Horsepower
Brake Mean Effective Pressure, lb. per sq. in.
Fuel-Consumption, lb. per b.hp-hr.
Air Blast Speed, m.p.h.

FIG. 23—CURVES SHOWING THE VARIATION IN THE PERFORMANCE OF THE TYPE-K CYLINDER WITH THE BLAST SPEED

ent blast-directions. One set of curves was obtained with the air-blast directed against the exhaust side of the cylinder, which had a port-arrangement similar to that shown in Fig. 4 of the paper. The other set of points was obtained with the air-blast directed between the ports, at right angles to a plane through the two valves It will be noted that the flow of fuel could be reduced to a lower value with the blast directed between the valves, which is not in agreement with results obtained by the Royal Aircraft Establishment on the effect of blast direction. This discrepancy may have been caused by variations in the spark-plugs used during the tests of the type-K cylinder. In both cases the limit below which the flow of fuel could not be reduced was the point at which severe preignition occurred. This preignition probably was caused by the spark-plugs becoming so hot as to ignite the charge prematurely, and slight differences in the spark-plugs used in the two tests might account for the earlier occurrence of preignition with the exhaust-side air-blast. The power-output and the fuel economy obtained are considered very creditable for a cylinder of this size using cast iron as the head material. An interesting fact in connection with these performance results is that the cylinder operates in a reliable manner with temperatures at the hottest point on the head that have heretofore been considered prohibitive. It appears, however, that such temperatures are permissible provided the temperatures of the spark-plugs and

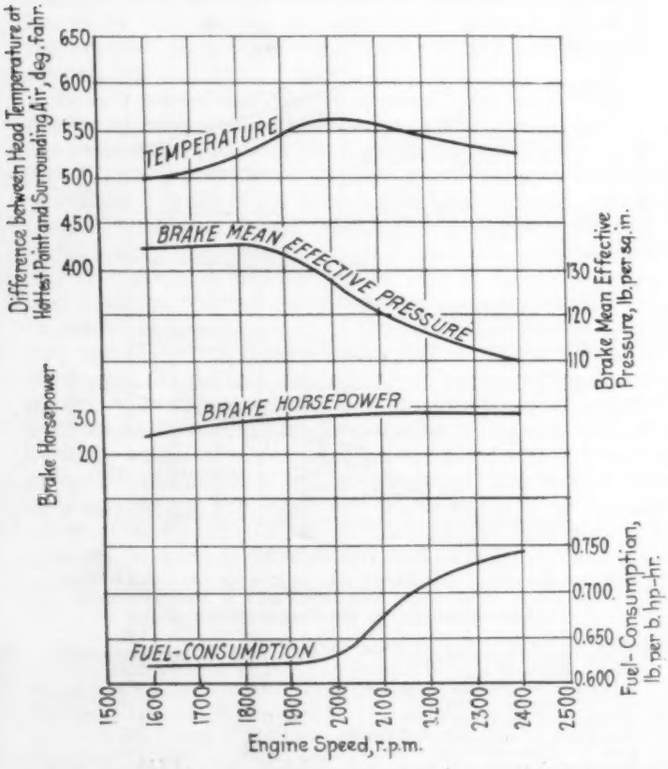
TEMPERATURE
BRAKE MEAN EFFECTIVE PRESSURE
BRAKE HORSEPOWER
FUEL-CONSUMPTION
Difference between Head Temperature at Hottest Point and Surrounding Air, deg. Fahr.
Brake Horsepower
Brake Mean Effective Pressure, lb. per sq. in.
Fuel-Consumption, lb. per b. hp-hr.
Engine Speed, r.p.m.

FIG. 22—VARIATION IN THE PERFORMANCE OF THE TYPE-K CYLINDER WITH THE SPEED

the exhaust-valves are not excessive. In the case of this cylinder, the exhaust-valve ran at a lower temperature than is experienced usually with most water-cooled designs with normal water-temperatures. The spark-plugs were the best that could be obtained from the standpoint of resistance to preignition. This same cylinder has run without failures for nearly 50 hr. at wide-open throttle with head-temperatures of over 500 deg. fahr.

Fig. 22 shows the variation in performance with speed of the same cylinder with the air-blast directed against the exhaust side. The valve-timing of this particular cylinder was designed for a normal speed of 1800 r.p.m., which accounts for the considerable decrease in mean effective pressure at the higher speeds. As shown in Fig. 21, the fuel-consumption of 0.64 lb. per b. hp-hr. at the normal speed of 1800 r.p.m. could have been reduced considerably by the use of a leaner carbureter-setting. The change in the head-temperature with the speed of rotation is small, which is typical of air-cooled cylinders of this type. The high fuel-consumption and the high blast-speed shown by this set of curves were used as a safety precaution on account of operating the cylinder at speeds so greatly in excess of those for which it was designed.

Fig. 23 shows the effect of a reduction in the air-blast speed on the type-K cylinder, with the air-blast directed against the exhaust side. With the air-blast reduced to one-half its original value, the rise in the head-temperature was slightly over 100 deg. fahr., and the operation of the cylinder was still satisfactory with very little loss of power or increase in the specific fuel-consumption. However, on further reduction of the air-blast speed, preignition occurred.

EXHAUST-VALVE COOLING

The problem of exhaust-valve cooling has been discussed by Mr. Heron but, due to its primary importance in air-cooled cylinder work, it is of interest to mention some of the laboratory experiences in regard to this important problem. The exhaust-valves on many of the air-cooled cylinders developed by the Engineering Division run somewhat cooler than the exhaust-valves of a similar size in water-cooled cylinders, although the parts of the cylinder immediately surrounding the valves in every case run at a considerably higher temperature in the air-cooled cylinder. The cause of this apparently contradictory condition seems to lie in the fact that, in designing air-cooled cylinders, it has been realized that special attention to exhaust-valve cooling was necessary. The valves, guides and ports therefore were designed to afford the maximum opportunity for heat dissipation from the valve, and the valve itself has in most cases been designed with large areas for heat dissipation from the stem and the seat. This has resulted in valves that run cooler than valves of equal size in water-cooled cylinders that have had only ordinary attention to the design factors necessary for proper exhaust-valve cooling. The valve and port design that appears to give the best exhaust-valve cooling is that illustrated in Fig. 12 of the paper and developed personally by Mr. Heron. Just which of the features of this design is responsible in greatest measure for the excellent cooling of the valve has not yet been determined. The Engineering Division is planning to conduct a series of tests in the near future that will establish the basic factors of valve and port design that promote effective cooling of the exhaust-valve. The indications are that there is no fundamental reason why the exhaust-valve in an air-cooled cylinder should run at a lower temperature than a similar valve in a water-cooled cylinder, and it is believed that the differences in design account fully for the differences in the cooling that have been observed. It appears fairly certain that, if a design similar to that shown in Fig. 12 were used in a water-cooled cylinder, the temperature of the valve would be even lower than in the case of an air-cooled cylinder, under similar conditions of power-output and mixture-strength.

With further reference to the problem of valve cooling, it should be remembered that excess fuel or oil in the combustion-chamber has a very marked effect on the temperature of the exhaust-valve. It is very easy for the experimental engineer to lose sight of this fact and attribute an improvement in valve cooling to changes in the design, when the lower temperature is merely due to a richer mixture or a larger amount of lubricating oil passing the piston. In comparing exhaust-valve temperatures, it is absolutely necessary to hold the mixture-ratio and the amount of oil reaching the combustion-chamber as nearly constant as possible, since these two factors may have a more powerful effect than any changes in the design of the valve or the port. A very badly designed exhaust-valve can be made to run at comparatively low temperatures by using an excessively rich mixture or an excessive amount of lubricating oil.

INTERNALLY COOLED VALVES

The Engineering Division has done a considerable amount of experimenting with internally cooled valves; that is, valves with hollow stems filled with certain high-boiling-point liquids. The object of such arrangements is to afford a rapid conduction of heat from the head of the valve to the upper portions of the valve-stem, thus causing a higher rate of heat-transfer due to the greater temperature-difference between the stem and the valve-guide. This method of valve cooling is fairly effective and often adds very little to the cost of the valve since, in many cases, a hollow-stem valve must be used, whether or not it is filled with liquid, to provide the necessary stem-surface for the proper heat-conduction without excessive weight. It is probable that internal valve-cooling will be found unnecessary for anything but very large air-cooled cylinders required to run for long periods at full throttle, since careful attention to the design will make internal cooling unnecessary in water-cooled cylinders and in air-cooled cylinders in which the valve conditions are less severe.

AIR AND WATER-COOLED TYPE COMPARISONS

In concluding Appendix 1, it is of some interest to review the relative advantages and disadvantages of the air-cooled and water-cooled types of engine from the point of view of the not inconsiderable experience of the Engineering Division with both types. It seems almost certain that the water-cooled engine will be shorter and more compact, due to the smaller space required between the adjacent cylinders, and particularly if it is conceded that the radiator may occupy space not required for other purposes with either type of engine. It appears evident also that the water-cooled engine will have the advantage of convenience due to the removable head, since air-cooled design has not yet reached the stage where the removable cylinder-head is entirely practicable. The water-cooled engine probably will have a slight advantage in regard to silence of operation also, provided equal care in the design is used for both types. This is on account of the fact that the block-cylinder construction lends itself more readily to the enclosed valve-gear, and the water-jackets seem to form a fairly effective barrier against excessive

FIG. 24—THE TYPE-H CYLINDER-HEAD PRODUCED BY DOUBLE POURING

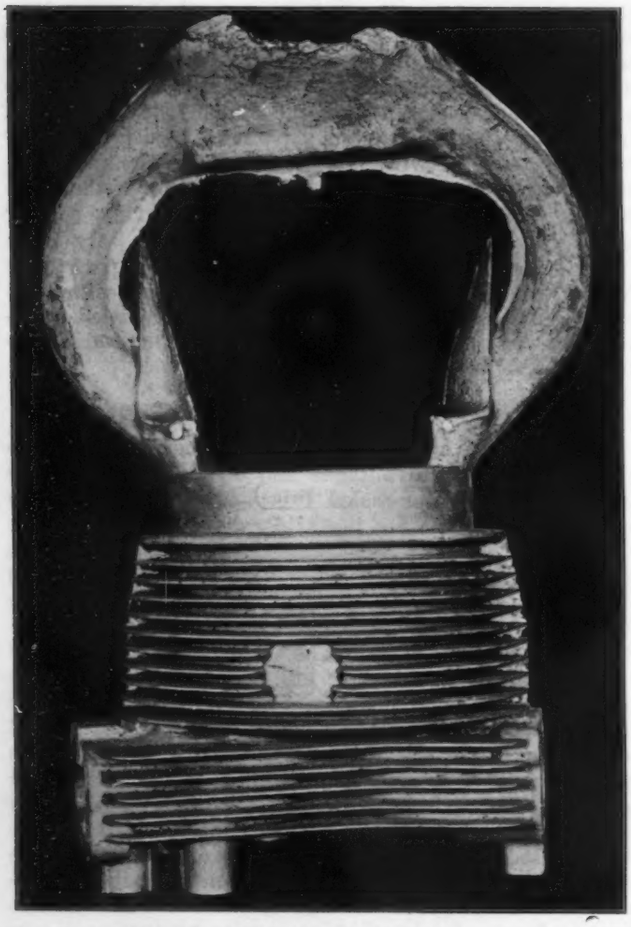

FIG. 25—THE TYPE-I CYLINDER-HEAD PRODUCED BY SINGLE POURING

noise. Perhaps the greatest advantage of the water-cooled engine lies in the fact that, for a given grade of fuel, a higher compression-ratio and a leaner mixture can be used, which naturally results in a somewhat more economical engine. While the advantage of the water-cooled engine in this respect is greatest at full throttle and becomes less important for engines required to operate largely at throttled speeds, it warrants consideration where fuel economy is a highly important factor. This fact conflicts with the general opinion of the motoring public that the air-cooled automobile is inherently more economical than the water-cooled vehicle, which opinion has undoubtedly resulted from the fact that the best known American air-cooled car has been designed with factors outside of the engine that result in a high economy when measured in miles per gallon.

The air-cooled engine, on the other hand, scores heavily from the point of view of convenience of handling, especially under extreme weather conditions, either of heat or of cold. The wide temperature-range under which an air-cooled engine will give satisfactory operation adapts it for service under very severe conditions and the freedom from freezing and boiling is an item that every motorist can appreciate. In the well designed air-cooled vehicle, the cooling system should require no attention whatever for months at a time. Other cooling-system troubles, characteristic of the water-cooled engine and absent in the air-cooled engine, are those due to formation of scale in the water-jackets and the ever-present leakage from radiators, hose connections and water-pumps. The formation of scale is a factor of more importance than is commonly supposed and if heavy, may result in definitely poor operation of a water-cooled engine at maximum power-output. Table 4 illustrates the effect of a small amount of scale on the wall temperatures of a water-cooled cylinder.

TABLE 4—EFFECT OF SCALE FORMATION ON CYLINDER-WALL TEMPERATURES[14]

Temperature, Deg. Fahr.	No Scale	Scale	Thickness of Scale, In.
Water Outlet	171	171	....
Cylinder Barrel	182	181	....
Combustion-Chamber Wall	253	292	0.012
Exhaust-Valve Seat	241	316	0.023
Head between Valves	307	391	0.025

[14]Water-cooled 5 x 7-in. cylinder run at full throttle at 1700 r.p.m. with constant cooling-water temperature. Compression ratio, 5.5 to 1. Fuel used, 20 per cent of benzol and 80 per cent of aviation gasoline.

The air-cooled powerplant also should be somewhat lighter in weight than the water-cooled powerplant since, in spite of the added length of the engine and the addition of cooling fins, fan and air-ducts, the elimination of water-jackets, radiator, pump, piping and water constitutes a saving in weight that will result in an overall saving for the air-cooled powerplant in most cases. The individual removable cylinders of the air-cooled engine are a convenience in overhauling that largely offsets the convenience of the removable head of the water-cooled engine.

CHOICE OF POWERPLANT TYPE

Without going into the question in detail, it appears that the well designed air-cooled and water-cooled powerplants are approximately equal in cost, ease of production, smoothness and flexibility of operation, and road performance, with cars of equal weight and engines of equal displacement.

In view of the above considerations, it is apparent that

the choice of the type of powerplant will depend on the individual conditions under which the vehicle is to be operated. It is the opinion of the writer, however, that

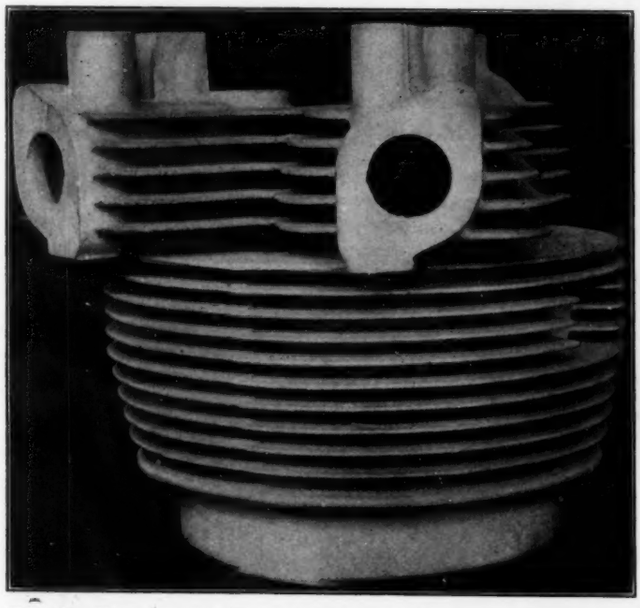

FIG. 26—VIEW OF THE TYPE-I CYLINDER-HEAD SHOWING THE PORTS

the advantages of the air-cooled engine from the standpoints of convenience to the driver and entire freedom from cooling-system trouble will result in a rapid and

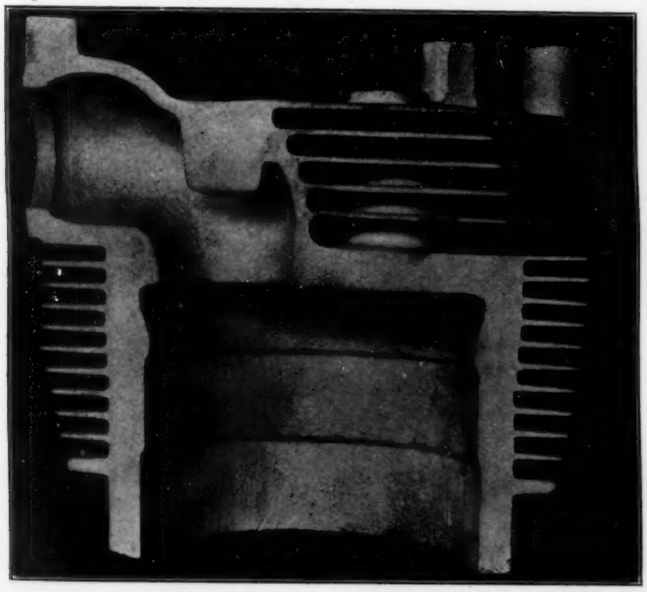

FIG. 27—SECTION OF THE TYPE-I CYLINDER-HEAD

healthy growth of this type of powerplant for automotive vehicles in this Country.

APPENDIX 2

FOUNDRY PRODUCTION OF AIR-COOLED CYLINDERS[15]

The production of air-cooled cylinder castings having fins cast integrally presents problems that are so different from those encountered in making water-cooled cylin-

[15] Contributed by E. H. Dix, Jr., in charge of the foundry and metallurgical branch, engineering division, Air Service, McCook Field, Dayton, Ohio.

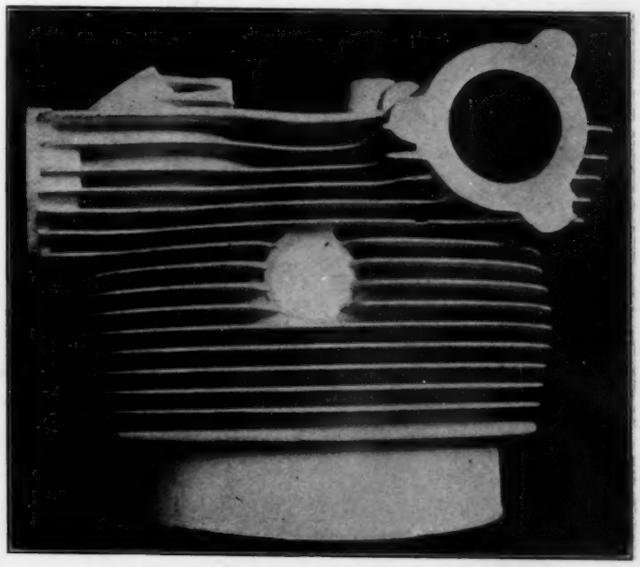

FIG. 28—THE TYPE-J TWO-PORT CYLINDER-HEAD

ders that there is much skepticism as to their ultimate practicability. The average engine-designer has had lit-

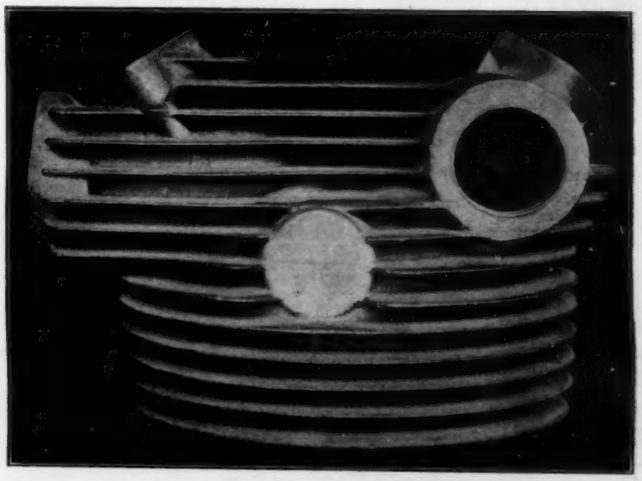

FIG. 29—THE TWO-PORT TYPE-K CYLINDER-HEAD

tle, if any, practical experience in the foundry. He is, therefore, more or less dependent upon the judgment and

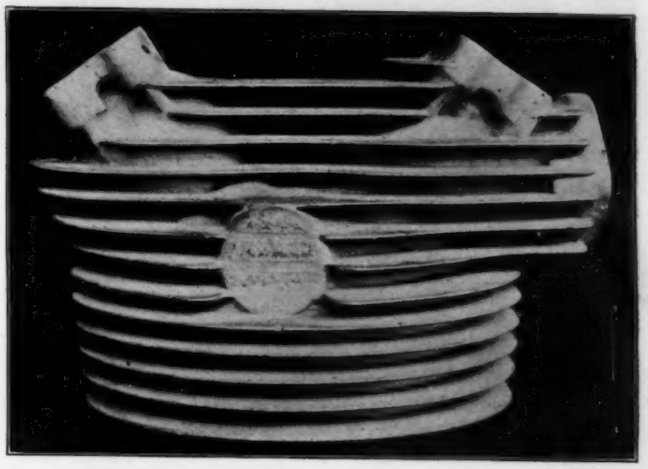

FIG. 30—ANOTHER VIEW OF THE TYPE-K TWO-PORT CYLINDER-HEAD

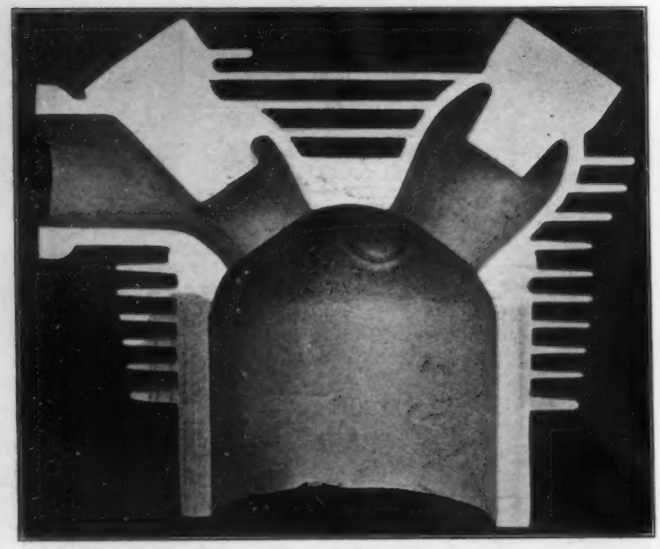

FIG. 31—SECTION OF THE TYPE-K CYLINDER-HEAD

advice of the foundryman. Foundrymen, as a class, are slow to adopt new ideas, and their judgments are apt to be based on past experience rather than future possibilities. This characteristic is evident from a study of the progress, or rather lack of progress, that has been made in foundry methods until recent years. The foundry has been the last manufacturing department to attract the attention of the technical man. Men learned the foundry trade by growing up in a foundry, a hard and tedious method; but, once acquired, knowledge was possessed which was carefully guarded, especially from men in the other branches of the industry. This policy or habit retarded advancement and there was little exchange of ideas. It is not surprising, then, that the foundryman would advise against attempting to make castings with such long thin fins as closely spaced as is required by efficient design. Also, after having once given such advice, it is natural that, where such designs were attempted, they failed due to foundry difficulties. It merely proved that the foundryman's judgment had been correct; but, as an actual fact, his cooperation had been only half-hearted; he had not expended his best efforts. However, if it is once proved to him what can be

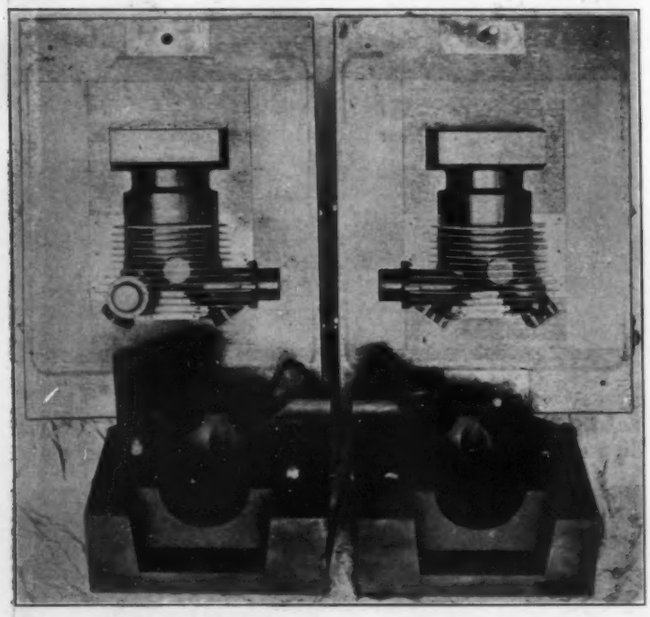

FIG. 32—THE PATTERN EQUIPMENT USED TO PRODUCE THE TYPE-K CYLINDER-HEAD

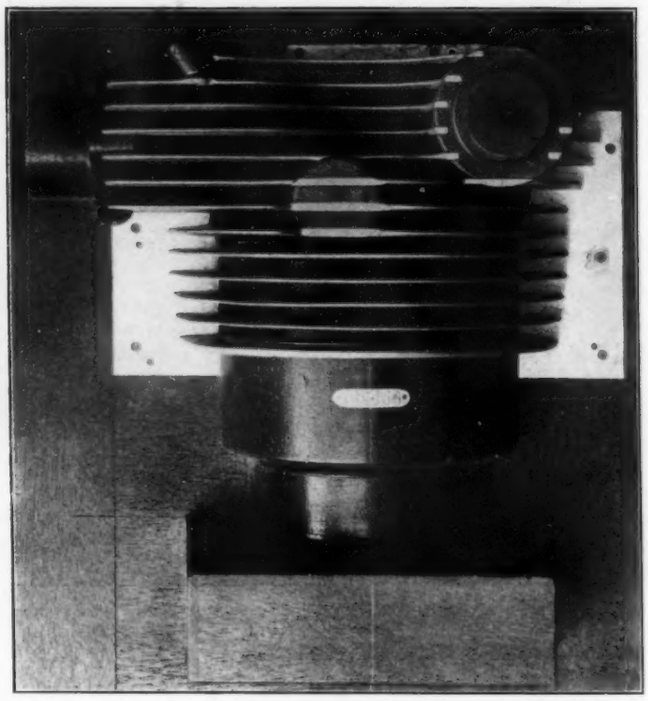

FIG. 33—A DETAILED VIEW OF THE PATTERN PLATE FOR THE DRAG PORTION OF THE TYPE-K MOLD

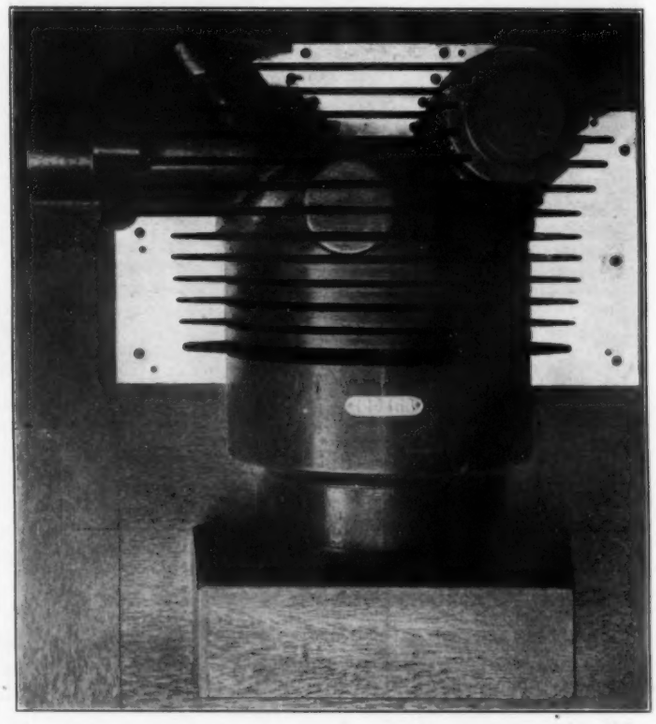

FIG. 34—THE PATTERN PLATE FOR THE TYPE-K CYLINDER-HEAD WITH THE SHEET-ALUMINUM FINS REMOVED

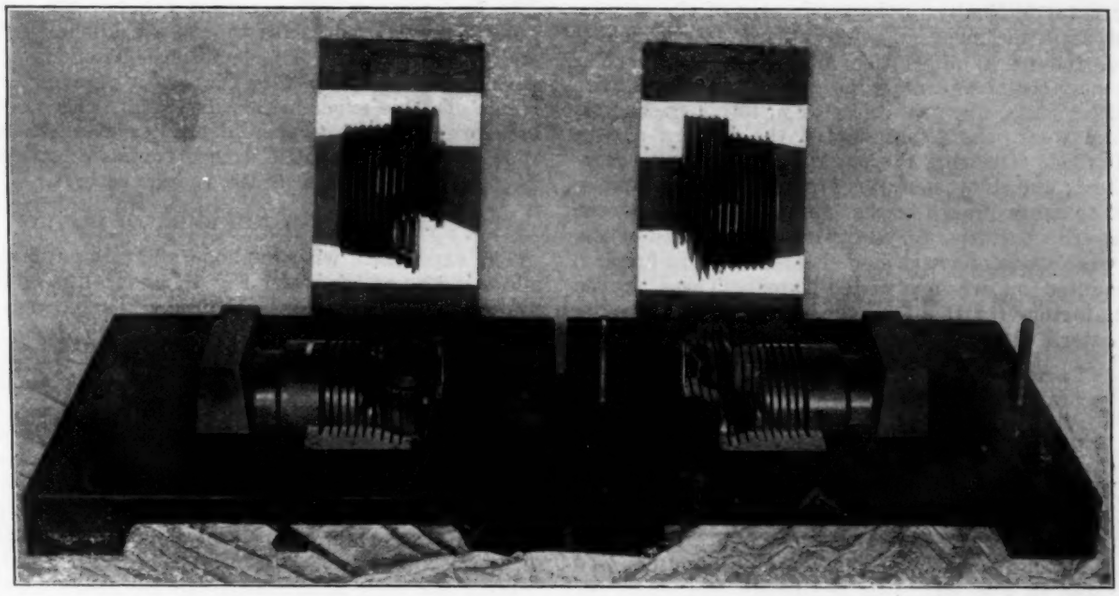

FIG. 35—VIEW OF THE TYPE-K PATTERN EQUIPMENT SHOWING HOW THE SHEET-ALUMINUM FINS ARE MOUNTED

done, he will not be content until he has accomplished corresponding results.

CASTING ALUMINUM-ALLOY CYLINDER-HEADS

It is hoped that a discussion of the experience of the Engineering Division foundry in casting aluminum-alloy cylinder-heads for air-cooled engines will stimulate interest and encourage the development of air-cooled motor-car engines. The various types of cylinder-head casting made to date are illustrated in Figs. 24 to 31. Both the four-port type-H shown in Fig. 24 and the three-port type-I castings illustrated in Figs. 25 to 27 require outside cores, and this is a disadvantage where accurate smooth fin-surfaces are required. The two-port type-J casting shown in Fig. 28 is an improvement in that only a cover core for the intake port is required. This can be eliminated by having the lugs made as loose pieces to draw in toward the center.

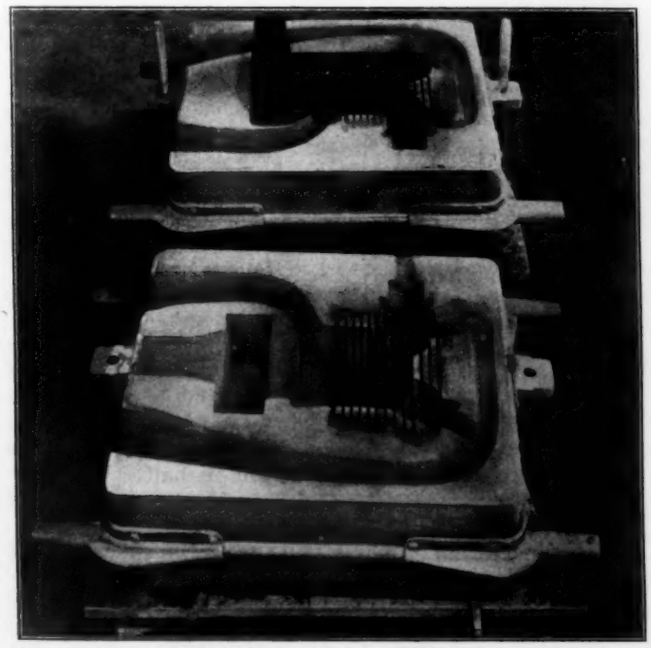

FIG. 36—THE TYPE-K MOLD AND CORE SHOWING THE METHOD OF GATING

The two-port type-K castings illustrated in Figs. 29 to 31 require no outside cores, and the body core is made in one piece. This cylinder-head is a triumph of design from the foundry viewpoint, and shows what can be accomplished in the way of simplifying the foundry processes if due consideration is given that point in the original layout.

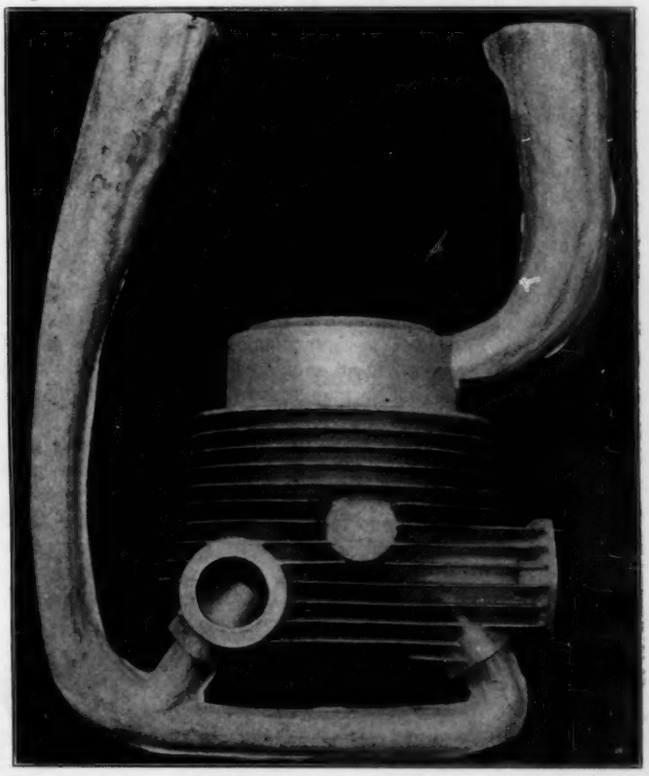

FIG. 37—THE TYPE-K CYLINDER-HEAD CASTING WITH THE GATES AND THE RISER ATTACHED

The greatest difficulty in the production of these cylinder-heads is in the molding. Once a satisfactory mold is prepared, the metal difficulties are few; and, for this reason, much emphasis must be placed on the pattern equipment. Metal fins are a necessity; although, for experimental work, the remainder of the pattern can be made of wood. Drawing the pattern is a difficult operation at best; and, if wooden fins are used, the warpage is sufficient to cause breaks in the tongues of sand between the fins. The patterns for the first three cylinders described were made of wood, had sheet-aluminum fins and were mounted on wooden match-plates. These proved very satisfactory for the development stage.

The pattern equipment for the two-port type-K cylinder is shown in Fig. 32. The simplicity of the core-box deserves comment; the core is made in one piece and thus eliminates all chance for core-shifts and lack of uniformity due to pasting cores. Both plates are constructed so that the fins strip through the pattern, as is brought out by the detailed view of the pattern plate for the drag in Fig. 33. Fig. 34 shows the fins removed. The patterns and plates are of wood, although the plates are reinforced by sheet aluminum. The fins are sheet aluminum mounted as illustrated in Fig. 35. This equipment was made for hand operation and has proved to be very successful, except that, after considerable use, the fit of the fins in the pattern has become loose in some places, which allows thin points of sand to be rammed between the fin and the pattern. These must be washed off to prevent dirty castings. An all-metal pattern for production work would obviate this trouble. By stripping the fins through the pattern-board, the drawing of the pattern becomes a simple operation and most of the molding difficulties are overcome. This method is believed to be more satisfactory than the joint stripper-plate.

The type-K cylinder with the pattern arrangement illustrated in Fig. 35 is well adapted to machine-molding except for the substitution of metal in place of wood. Either a jolt stripper or a jolt and roll-over machine can be used. If used on a jolt and roll-over machine, the plate to which the fins are secured would be attached to the draw-arm and the other plate carrying the pattern would be hung from the first plate by close-fitting pins which would not engage until the fins had been withdrawn 1 in. or more, thus accomplishing two results in one operation.

In casting the aluminum cylinder-heads previously illustrated, dry-sand molds have been used exclusively. Sand from the ordinary heap, strengthened by molasses water, has been found to be satisfactory. Facings are unnecessary and lead to complications. Both halves of the mold are sprayed with molasses water, and the core is set in the drag. The two halves are then placed separately in a core-oven and baked at about 400 deg. fahr. for 6 hr. The mold is then closed and poured while hot, as this aids in running the fins. Recently, attempts have been made to cast the type-K cylinder-head in green sand. These experiments seem to indicate that the green-sand mold is not suitable for this type of work. To run the fins in green sand, it is necessary to pour at a higher temperature and much faster than in the case of dry sand. The fast pouring often causes the narrow tongues of sand to wash, and seems to cause cold-shuts where the fins join the body. The metal has a tendency to blow when poured rapidly into the green-sand molds. As a result of these experiments, dry-sand molds are strongly recommended. The splendid success attained with this type of mold gives little excuse for making these heads in cores that give inaccurate, rough castings.

GATING FOR ALUMINUM-ALLOY CASTING

The experience with methods of gating for aluminum alloys is also of interest. Fig. 24 shows how some of the first cylinder-heads were poured. These were poured two-up, and the arrangement was such as to get the metal into the mold as quickly as possible. Although it might be argued that this method is likely to cause the mold to wash and also to cause drossy metal, no difficulty was experienced in this connection. Fig. 25 illustrates a similar scheme, except that single instead of double pouring was used. A dirt-trap ring above the cylinder was added as an extra precaution. A pouring temperature of 1425 deg. fahr. was used with the methods of gating already described.

With the type-K cylinder, bottom-pouring was tried and found to be very successful. Fig. 36 shows the open mold and illustrates this method of gating. The casting with the gates and the riser attached is illustrated in Fig. 37. This casting was poured at 1460 deg. fahr.

It had been customary to take off three rows of vents from each fin in both the cope and the drag, to allow easy escape of the gas and to prevent short-run fins. However, no vents were taken off the fins of the casting shown in Fig. 37, and the fins were full-run and much smoother than when vents were taken off. The elimination of the vents will save much time in the molding, and is particularly desirable for machine-molding where the operator is not a skilled molder. Another thing that is being considered is the number of wires required to support the sand between the fins. For the hand-rammed dry-sand molds, three rows of wires have been used in both the cope and the drag. However, with machine-molding, it is believed that this can be reduced to one row or perhaps eliminated altogether, except in one or two very deep draws. This will be a very considerable item in production.

The mold shown in Fig. 36 was completely prepared by hand in about 2 hr. by the use of the stripper-plate pattern illustrated in Fig. 35, although it was necessary to wash the edges between each fin, due to the fine fins of sand caused by looseness in the fit of the fins in the pattern. By the elimination of this loss of time and by having the gates and the risers included on the boards, the time of molding could easily be cut in half. With machine-molding, a production of 8 to 10 per hr. might be obtained easily.

It is hoped that this brief outline of the experience of the Engineering Division foundry will show that air-cooled cylinder-head castings of the types illustrated are far from being bad production propositions. They are, in fact, much less difficult than some of the water-cooled designs requiring complicated and intricate jacket-cores.

Vol. XII January, 1923 No. 1

Cooling Capacity of Automobile Radiators

By E. H. Lockwood[1]

Annual Meeting Paper *Illustrated with* Charts

THE heat-dissipating properties of three types of radiator core have been investigated at the Mason Laboratory, Yale University. These include the fin-and-tube, the ribbon and the air-tube groups, so classified according to the flow of the water and the air. The ratio of the cooling surface to the volume is shown to be nearly the same in the fin-and-tube and the air-tube cores, while that of the ribbon core is somewhat greater. A formula is derived for computing the heat-transfer coefficient, which is defined as the number of heat units per hour that will pass from one square foot of surface per degree of temperature-difference between the air, and the water and is the key to radiator performance, as by it almost any desired information can be obtained. When the heat-transfer coefficients have been found for a sufficiently wide range of water and air-flows the cooling capacity of a radiator can be computed for any desired condition. In an appendix five fundamental and six derived formulas are given. These cover such topics as pump circulation, thermo-syphon circulation, limiting depth of core required, final temperatures, weight of water-flow and the like. The results of applying the formulas to six cores that were selected as representative of the three main types are illustrated by a series of charts.

A RADIATOR is used to cool the jacket-water of engine cylinders. Its ability to dissipate heat depends not only on the extent and form of its cooling surface, but also on the velocity at which the air and the water flow past the surface. The air velocity in the case of automobiles and trucks is frequently low, and must be supplemented by a fan driven by the engine.

The value of a radiator is determined by other factors than its ability to dissipate heat, but this alone will be considered in the present discussion which will describe a method of computing the cooling capacity of radiators, under any reasonable operating conditions, that is based on coefficients determined by laboratory tests. An application of the method has been made to several types of radiator and their cooling characteristics have been disclosed.

Classification of Radiators

A radiator contains tanks located on the top and at the bottom to receive and to distribute the water. A cellular portion, usually called the core, occupies the space between the top and the bottom tanks and serves to cool the water. Numerous small passages allow the water to flow through the core, while provision is made for air to circulate freely around the hot metal to remove the heat. In addition to the parts mentioned, a protecting case usually is added to support the radiator and to give a pleasing appearance to the front of the vehicle.

The core is made preferably of rectangular outline and of uniform depth or thickness. It is constructed in a variety of forms, but all have the same characteristic thin-walled water-passages, with free air circulating on their exterior. With regards to their heat-dissipating properties, cores may be divided into three classes: (*a*) fin-and-tube, (*b*) ribbon and (*c*) air-tube.

In the first type the water passages are straight tubes leading from the top to the bottom tanks, with fins attached to give an increased surface for removing the heat. The tubes usually are cylindrical, but occasionally are drawn in oval or flat forms. Disc fins are made by soldering to the tube at close intervals round or square metal plates of about twice the diameter of the tube. This method of construction is varied by using, instead of fins, strips that are punched to receive a row of tubes. A more common arrangement is called the continuous fin, since it consists of a larger plate punched to receive all the tubes of the core. Whatever the type may be, the fins are spaced about equal distances apart and are usually five or six per inch. The tubes are of ¼ to ⅜-in. diameter.

In the ribbon-core type the water passages are formed between parallel plates, which are separated slightly by crimping the edges, or by using a spacing wire and then soldering them. The water ribbons are made the full depth of the core, and extend from the top to the bottom, where they are fastened to the water-tanks. The ribbons have either straight or zigzag channels. They are separated at regular intervals by the attached ribs, or fins, which serve the double purpose of stiffening the core and increasing the amount of surface for dissipating the heat. The attached fins are laid out in regular lines and give to the front a cellular appearance of squares, diamonds or hexagons, according to the shape of the ribbons. Two sides of each air-passage as a rule are bounded by water ribbons and the others by fin surfaces. It is possible, however, to join the corners of zigzag ribbons together to form a core without connecting the fins, and this construction is sometimes used. The water passages in the ribbons are from 0.05 to 0.06 in. wide. The air cells are from 3/16 to ⅜ in. square.

Air-tube cores are made up of short tubes packed closely together in horizontal rows, with their ends flared and soldered to form the front and the back of the core. Air passes through the tubes, while the water fills the entire space around the tubes, and flows in any path from the top to the bottom. In appearance air-tube cores resemble ribbon cores; and both belong to the cellular class. Air tubes have no fins, the surface being heated by direct contact with the hot water. In some cases baffles, or dents, are formed in the tubes to increase their effectiveness in cooling. This result is produced, not by an increase of the surface, but by a turbulence in the air-flow that renders the same amount of surface more effective. Air-tube cores have been made with both round and square tubes, but neither type has been much used owing perhaps to the cost of manufacture. Diagrammatic drawings of all three types of core are shown in

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Fig. 1. The number of examples under ribbon cores is so great that only a few common forms are illustrated.

Measurements of Radiator Cores

The cooling capacity of a radiator depends chiefly on the core, since the water-tanks have relatively little effect. Correct measurements of the core are therefore of importance in determining its capacity. The size of a radiator is expressed by the frontal area and the depth of the core. The combined area of the air passages through the core, called the free-air area, is a factor in its performance.

Average values of the free-air area of typical cores, expressed as percentages of the frontal area, are given in Table 1. It will be seen that the values range from 65 to 75 per cent for all cores; hence the free-air area is a fairly large percentage of the frontal area, and is nearly constant for all types of core.

The capacity of a radiator is closely connected with the total amount of surface exposed to the air currents. This measurement, like the free-air area, is determined by the mensuration of the core, and is usually expressed in two parts. The surface enclosing the water channels is called the direct surface, and is assumed to be more effective, since it is in direct contact with the hot water. The remaining surface, consisting of fins or ribs, receives heat by conduction and is called the indirect surface. When the term surface is mentioned without qualification, it refers to the total or sum of the direct and the indirect surfaces. The surfaces of different cores can be compared only when they have the same volume. The results of comparing several types of core are given in Table 1. The unit of comparison has been the surface of a core having a volume of 1 cu. ft. Fin-and-tube cores have about the same amount of surface as air-tube cores, or from 110 to 130 sq. ft. for each cubic foot of volume. Certain ribbon cores have about the same amount of surface, but some small-sized cell

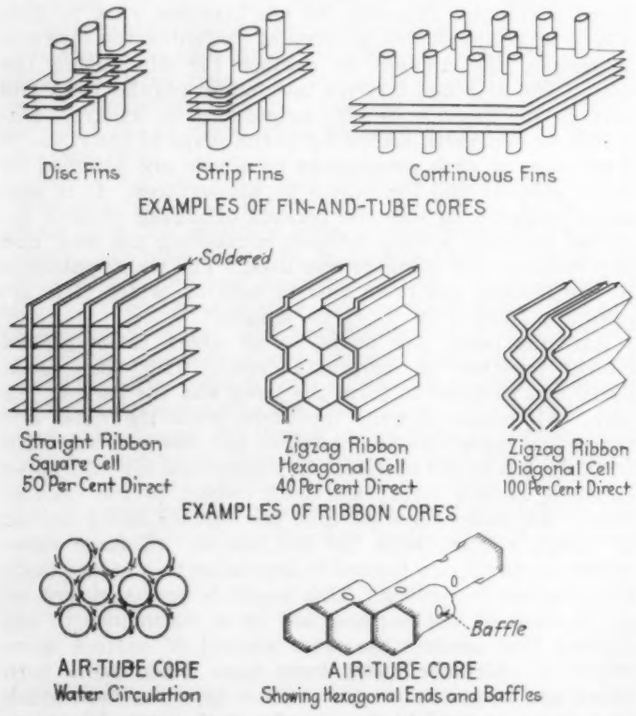

Fig. 1—Examples of the Three Types of Radiator Cores in General Use

Table 1—Average Values of Free-Air Area and Total Core Surface

Type of Core	Free-Air Area in Terms of Frontal Area, Per Cent	Direct Surface in Terms of Total Core Surface, Per Cent	Total Core Surface, for 1 Cu. Ft. of Volume, Sq. Ft.
Fin-and-Tube Disc Fins	57	16	110
Fin-and-Tube Continuous Fins	70	15	130
Ribbon Core	65 to 80	50	150 to 175
Ribbon Core	65 to 80	100	125
Air-Tube Core	72	100	110

cores with straight ribbons have a unit surface as large as 175 sq. ft.

The percentage of direct surface for the different types of core is given in Table 1. As stated above, it consists of a surface in contact with the water. The direct surface varies widely, being 100 per cent for air-tube cores and only 15 per cent for fin-and-tube cores. The values given in Table 1 are to be considered as general averages only. For use in computing any particular core the free-air area and core surface must be carefully measured.

Experiments on Cooling Capacity of Radiators

The heat dissipated by a radiator can be measured by supplying a known quantity of hot water at a constant temperature, and observing the drop of the temperature of the water for different air-velocities through the core. The apparatus requires some form of wind-tunnel, with means for adjusting the rate of air-flow. It also requires an ample water-supply at a constant temperature with means for metering and controlling the rate of flow. The experiments described were made on apparatus of this kind, where direct measurements were made at all temperatures, velocities, and the like. The air velocity was not measured directly, however, but was computed from the air temperatures by the thermal relations.

Comparison of the cooling capacity of two radiators requires identical air-velocities and identical temperatures of the entering water and the air. Since it is practically impossible to maintain these conditions constant, to make a comparison it is necessary to make a final reduction to a common basis. Formulas for this coefficient to the other radiator test-data are given in the Appendix. The starting point for this theory was the assumption that the rate at which heat leaves the surface is proportional to the difference in temperature between the metal and the air. The defect of the theory is that it takes no account of the effect of the velocities of the air and the water; hence these two variables must be dealt with independently, as will be shown in the next paragraph.

Heat-Transfer Coefficients

The theory of the transmission of heat from water to air through a thin metal wall involves the rate at which the heat is transferred from the wall or surface. The rate of transfer is expressed by the heat-transfer coefficient which is defined as the number of heat units per hour that will pass from one square foot of surface per degree of temperature-difference between the water and the air. In the case in which the temperature of both the water and the air changes in passing through the radiator, the arithmetical mean of the entrance and exit temperatures is taken. The relation of the heat-transfer coefficient to the other radiator test-data is given in the appendix, where formula (1) is used for computing the coefficient.

When a radiator is tested, a series of runs is made covering the desired range of air and water velocities, for each of which the heat-transfer coefficient is computed. The radiator performance is then estimated, not from the tests, but from the computed quantities found by the aid of the coefficients. It is desirable to plot the coefficients on cross-section paper so that inconsistent or erroneous results can be detected and the intermediate values interpolated.

An example of the heat-transfer coefficients for a ribbon core is given in Fig. 2. It is apparent that the coefficients are not constant, but increase with both the water and the air velocities. Similar diagrams have been made for other types of radiator, for each of which values have been interpolated for the same velocities, forming the new diagram shown in Fig. 3.

Six cores have been chosen as fairly representative of the three main types. The effect of high and low air-velocities is clearly shown by the right and the left sides of the diagram. At low air velocity the air-tube and the

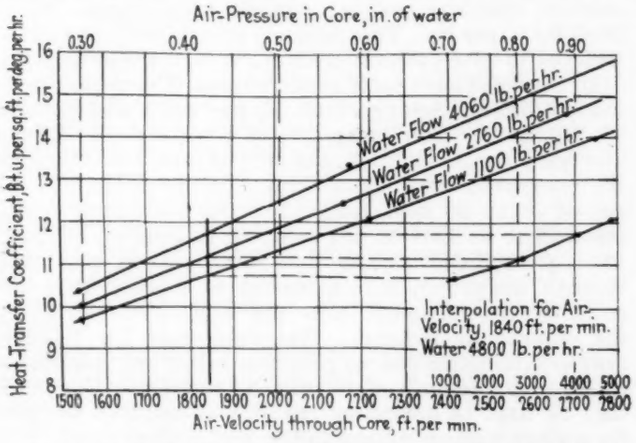
Air-Pressure in Core, in. of water
Heat-Transfer Coefficient, B.t.u. per sq. ft. per deg. per hr.
Water Flow 4060 lb. per hr.
Water Flow 2760 lb. per hr.
Water Flow 1100 lb. per hr.
Interpolation for Air-Velocity, 1840 ft. per min.
Water 4800 lb. per hr.
Air-Velocity through Core, ft. per min.

Fig. 2—Heat-Transfer Coefficients Showing the Variation Produced by Changes in the Flow of Air and Water

ribbon cores are about on a par. At high air-velocity the air-tube surface is more effective than the ribbon core. In both cases the fin-and-tube surface is inferior to either of the others.

Comparison of Cooling Capacity

When the heat-transfer coefficients have been found for a sufficiently wide range of water and air-flows, the cooling capacity of a radiator can be computed for any given condition. It is scarcely too much to say that the coefficients are a key to the radiator performance, for by their aid almost any desired information can be had. A useful expression for cooling capacity is given in the appendix, formula (6). It requires as given data: the initial water-temperature, t_1; the initial air-temperature, t_2; the rate of water-flow, w; the rate of air-flow, W; the core surface, S; and the heat-transfer coefficient, k. The air-flow rate must be computed by formula (2), using the assumed air-velocity and free-air area of the core. In comparing the cooling capacity of the two radiators, it is necessary to assume the same frontal-area and core-depth, as well as equal temperatures and velocities.

The foregoing method has been applied to the six cores whose heat-transfer coefficients were given in Fig. 3. The cooling capacities are given in Fig. 4, where the results are arranged in the same general way as in the preceding diagram. In comparing the two diagrams, it must be remembered that the heat-transfer coefficient takes account only of the quality of the surface for heat

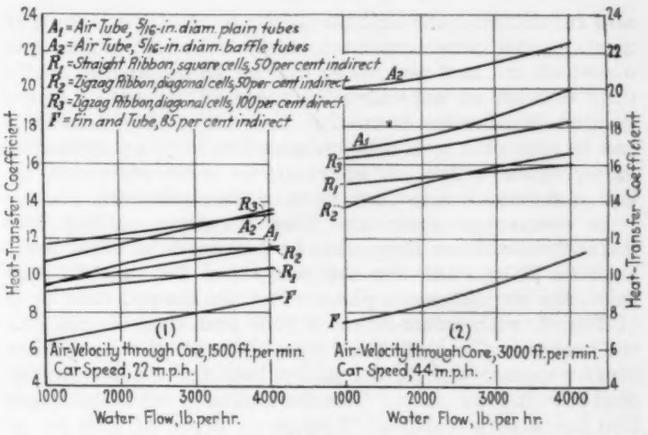
A1 = Air Tube, 5/16-in. diam. plain tubes
A2 = Air Tube, 5/16-in. diam. baffle tubes
R1 = Straight Ribbon, square cells, 50 per cent indirect
R2 = Zigzag Ribbon, diagonal cells, 50 per cent indirect
R3 = Zigzag Ribbon, diagonal cells, 100 per cent direct
F = Fin and Tube, 85 per cent indirect
Heat-Transfer Coefficient
(1) Air-Velocity through Core, 1500 ft. per min. Car Speed, 22 m.p.h.
(2) Air-Velocity through Core, 3000 ft. per min. Car Speed, 44 m.p.h.
Water Flow, lb. per hr.

Fig. 3—Comparative Heat-Transfer Coefficients for Different Cores

exchange, but not its extent. On the other hand, the capacity diagram takes account of both the quality and the extent of surface of each core; hence it is a true measure of its cooling ability.

Fig. 4 suggests several interesting conclusions, which may be stated as follows:

(1) As between ribbon and air-tube cores as classes, there is little to choose for the speeds used in automobiles. For low air-velocity, ribbon cores are slightly better. For higher air-velocity the results are about on a par

(2) The baffles, or dents, used in air-tube cores add materially to the cooling, without increasing the surface or weight

(3) The straight-walled ribbon core, having small square cells with 50 per cent of indirect surface, cools as well as other ribbon cores with 100-per cent direct surface. This result is partly due to the greater amount of surface possessed by the small cell core, but it also indicates that the indirect surface of this type of core is nearly equal to the direct surface in cooling ability

(4) The fin-and-tube core has only about three-quarters of the cooling capacity of the best forms of ribbon core of the same size. This is not surprising in view of the large amount of indirect surface contained in the fin-and-tube type

Thermo-Syphon Cooling

That the capacity of a radiator is greatly lessened by reduced water-flow has been shown clearly by Fig. 4, where the rates of flow vary from 1000 to 4000 lb. per hr. The heat exchange in thermo-syphon cooling can be computed by formula (7) of the appendix. This formula differs from formula (6) by omitting the water-flow, w,

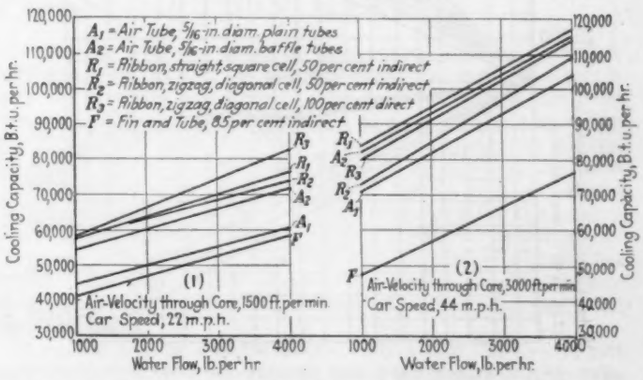
A1 = Air Tube, 5/16-in. diam. plain tubes
A2 = Air Tube, 5/16-in. diam. baffle tubes
R1 = Ribbon, straight, square cell, 50 per cent indirect
R2 = Ribbon, zigzag, diagonal cell, 50 per cent indirect
R3 = Ribbon, zigzag, diagonal cell, 100 per cent direct
F = Fin and Tube, 85 per cent indirect
Cooling Capacity, B.t.u. per hr.
(1) Air-Velocity through Core, 1500 ft. per min. Car Speed, 22 m.p.h.
(2) Air-Velocity through Core, 3000 ft. per min. Car Speed, 44 m.p.h.
Water Flow, lb. per hr.

Fig. 4—The Cooling Capacity of Typical Radiator Cores

and substituting the exit temperature t_2. Otherwise it is used in the same way as formula (6), but it has one drawback in that the value of the heat-transfer coefficient depends on the water-flow; hence it cannot be interpolated in advance from the diagram. This drawback can be overcome by a trial computation with an estimated value, from which an approximate water-flow can be had, and from it a revised value of the coefficient.

In comparing pump and thermo-syphon cooling, the water-temperature drop can be assumed in each case, such as 200 to 185 for the pump and 200 to 150 deg. fahr. for the thermo-syphon. For the fin-and-tube core of Fig. 3, with water-flows of 1000 and 4000 lb. per hr., the heat-transfer coefficients may be assumed to be 6.4 for thermo-syphon and 8.7 for pump, both for an air-velocity of 1500 ft. per min. The fin-and-tube core in question has a core width of 19.00 in., a depth of 2.50 in., a frontal area of 2.24 sq. ft., a cooling surface of 62.00 sq. ft., and a free-air area of 1.57 sq. ft. From these values the weight of air, as found by formula (2), is $W =$ 10,200 lb. per hr. In solving formula (7) the air temperature is assumed to be 70 deg. fahr., and the water temperatures, as stated above, 200 to 185 and 200 to 150 deg. fahr.

For the pump, $H = 60{,}000$ B.t.u. per hr.
For the thermo-syphon, $H = 39{,}000$ B.t.u. per hr.

By formula (9) the water circulation found for each case is,

For the pump, $w = 4000$ lb. per hr.
For the thermo-syphon, $w = 780$ lb. per hr.

As the water-flows agree well enough with those assumed in estimating the heat-transfer coefficients, it is not necessary to choose new values, and the values of H may be assumed to be correct.

From these figures the cooling of thermo-syphon cores may be taken as about two-thirds of that for pump circulation, and the water circulation as about one-fifth as fast. These figures are based on an arbitrary temperature-drop in the radiator, and may be revised to suit the existing data. They are offered as showing a method of dealing with thermo-syphon circulation in a rational way. It is likely that the temperature-drop may be much less than that assumed, as an active circulation may be kept up by the steam-bubbles in the pipe leaving the cylinders, even with the water at nearly a constant temperature.

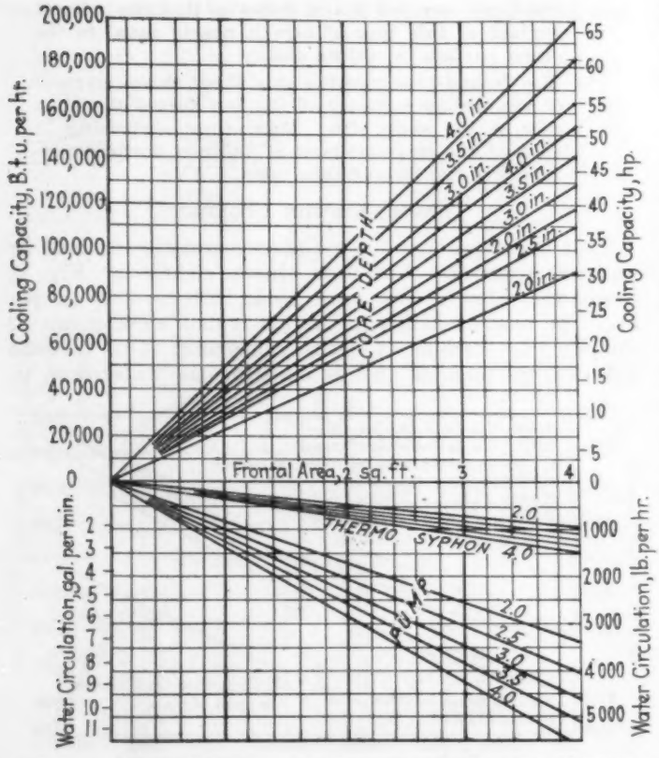

FIG. 5—CHART SHOWING THE COOLING CAPACITY FOR VARIOUS FRONTAL AREAS AND DEPTHS OF CORE

COOLING-CAPACITY CHARTS

The computed results for a radiator core may be put in convenient form for use as shown in Fig. 5. The frontal area is laid off on the horizontal center-line, with the cooling capacity above and the circulating water below. Each core-depth is marked on a sloping line, the heavy lines for the pump and the light lines for the thermo-syphon circulation. The cooling capacity is given in two units, the heat units on the left axis and the engine horsepower on the right. The two units are connected by the relation that the cooling required by 1 engine hp. is 3000 B.t.u. per hr. This figure is an arbitrary one but is believed to give sufficent cooling for automobile engines.

In constructing charts such as these it is necessary to assume limiting temperatures of the water and the air. In Fig. 5 the water temperatures were taken as 200 to 180 for pump circulation, and 200 to 140 deg. fahr. for thermo-syphon circulation. The air temperature was assumed to be 80 deg. fahr. The core air-velocity was assumed to be 1840 ft. per min., corresponding to about 25 m.p.h. Since all the lines are straight and pass through the origin, it is necessary to compute only one point by the formulas; thus the labor of constructing the diagram is reduced. The water rate for thermo-syphon circulation has no practical value, but was added to the chart because of its general interest. The chart may be used to determine the size of radiator suitable for a given engine, subject to the assumptions made. For example, a 50-hp. engine with pump circulation requires a core of 4-in. depth and 3-sq. ft. frontal area, and the engine pump must handle 9 gal. of water per minute.

LIMITING VALUES OF CORE-DEPTH

The cooling capacity of a radiator can be increased by adding to its depth, without changing the frontal area. The increased depth will be accompanied by a proportionate increase of the cooling surface and the quantity of circulating water, but no change in the air-flow. It follows that, although the capacity will increase with the depth, it will do so in a diminishing ratio, and will reach a limit where no further increase of the depth will be justified.

The limits of core-depth can be worked out from formula (8) which is given in the Appendix. In applying formula (8) it is necessary to assume initial and final values for the temperatures of both the water and the air. This can be done inasmuch as the water temperature-drop is to be constant for all depths, say from 200 to 180 deg. fahr., and the air temperature is to rise to a fixed limit, which may be put at 120 deg. fahr., from an initial temperature of 70 deg. fahr.

To illustrate, a ribbon core has a frontal area of 2.21 sq. ft.; a free-air area of 1.42 sq. ft.; a surface, for 1-in. depth, of 19.00 sq. ft. Tests of this core showed a heat-transfer coefficient, $k = 15.0$ for 2.875-in. depth, at a velocity of 2900 ft. per min. through the core.

The temperatures for use in (8) are: $t_1 = 200$ deg. fahr.; $t_2 = 180$ deg. fahr.; $t_3 = 70$ deg. fahr.; $t_4 = 120$ deg. fahr.; $M = 19$; $k = 15$. W is to be computed from

appendix formula (2), using the air-velocity and free-air area given above, whence $W = 17{,}800$ lb. per hr.

Substituting in (8), the value of d is found to be 7.9 in., which is the proper depth of core for the limiting conditions imposed.

The cooling capacity for the limiting depth is computed by formula (4) as 213,000 B.t.u. per hr. The required water-flow is given by formula (9) as 10,650 lb. per hr., or 22 gal. per min.

Deep cores are required only when high air-velocities are supplied. For example, in the core just described, if the velocity be reduced to 1800 ft. per min. with the corresponding value of $k = 13.0$, the depth is found by (8) to be $d = 5.7$ in.

In the preceding computations it has been assumed that the heat-transfer coefficient varies with the air and the water velocities, but not with the change in the depth of core. The coefficient probably varies slightly with the core depth; hence some error may be introduced by this assumption.

Radiators for Steam Condensation

In steam automobiles it has been found convenient to utilize the radiator to condense the steam, which is then available for feeding to the boiler, thus avoiding the frequent filling of the supply tank. The cooling capacity of the radiator when condensing the steam can be found in the same way as when cooling the jacket-water. It is necessary only to modify formula (7) by making $t_1 = t_2 =$ 212 deg. fahr. The value of the heat-transfer coefficient, k, is found as for hot water with the most rapid circulation. Radiators of the fin-and-tube type are most durable for steam where superheat is sometimes found in the exhaust.

The application of radiators to the condensing of steam can be illustrated by an example, in which the brake horsepower will be assumed to be 20. The steam consumption per hour will be about 20 x 15, or 300 lb. To condense this steam will require approximately 1000 B.t.u. per lb., or 300,000 B.t.u. per hr. It is pertinent to remark that if a 20-hp. engine had been used in a gasoline car, the heat removed from the jackets would be only 20 x 3000 or 60,000 B.t.u. per hr. This amount is only one-fifth of the heat removed in condensing the steam; hence it is safe to predict that the condenser for the steam car must be much larger than the radiator for the gasoline car.

In computing the size of radiator required, it will be well to select a considerable depth, say 5 in., for which the surface per square foot of frontal area will be 54 sq. ft., using the average values for fin-and-tube cores, Table 1. The free-air area may be assumed as 70 per cent of the frontal area and the air velocity through the area may be assumed to be 1840 ft. per min., corresponding to a speed of 25 m.p.h. The air-flow by formula (2) will be 5600 lb. per hr. The heat-transfer coefficient, k, according to Fig. 3 will be about 9.0.

Placing the above values in formula (7), the heat dissipated per square foot of frontal area is found to be 58,000 B.t.u. per hr. The frontal area required to condense the entire amount of steam will be 300,000 divided by 58,000, or 5.2 sq. ft., or the equivalent of 25 x 30 in. It will be interesting to compare these figures with the size of radiator used on the Stanley steam car in 1920, which was 23 x 23 x 5 in. Without doubt the condensing radiator of a steam car will be taxed more severely than the radiator for cooling the jacket-water of a gasoline car.

APPENDIX

A = the free-air area through the core
d = the depth of the core
H = the heat dissipated from the core
k = the heat-transfer coefficient computed by formula (1)
M = the surface of the core per inch of depth
S = the surface of the core
s = the specific volume of air at the mean temperature
t_1 = the temperature of the air entering the core
t_2 = the temperature of the water leaving the core
t_3 = the temperature of the air entering the core
t_4 = the temperature of the air leaving the core
V = the velocity of the air-flow through the core
W = the weight of the air circulating through the core
w = the weight of the water circulating through the core
0.24 = the specific heat of air at a constant pressure

Fundamental Formulas

$$k = [w(t_1 - t_2)] \div \{ S(\tfrac{1}{2}[t_1 + t_2] - \tfrac{1}{2}[t_3 + t_4]) \} \quad (1)$$

$$W = (60AV) \div s \quad (2)$$

$$H = w(t_1 - t_2) \quad (3)$$

$$H = 0.24\, W(t_4 - t_3) \quad (4)$$

$$H = \tfrac{1}{2}(t_1 + t_2 - t_3 - t_4)\, kS \quad (5)$$

The following formulas were derived from those given above by transformation

$$H = [0.48\, W(t_1 - t_3)] \div [1 + (0.48\, W/kS) + (0.24\, W/w)] \quad (6)$$

$$H = \{ 0.48\, W[\tfrac{1}{2}(t_1 + t_2) - t_3] \} \div [1 + (0.48 W/kS)] \quad (7)$$

$$d = [0.48\, W(t_4 - t_3)] \div [kM(t_1 + t_2 - t_3 - t_4)] \quad (8)$$

$$w = H/(t_1 - t_2) \quad (9)$$

$$t_2 = t_1 - (H/w) \quad (10)$$

$$t_4 = t_3 + (H/0.24\, W) \quad (11)$$

TAXATION ON A TIRE CONSUMPTION BASIS

MOTOR-VEHICLE taxation should be directly proportional to the inflicting of damage upon the road. Statistics should be compiled to determine where and why and how damage is done to the road, where true economy lies and where waste is created. A flat method of taxation for all cars, with the result that whether a car covers in one year 1000 or 100,000 miles, the tax on both would be identical in amount, should not be followed. The only way to redress this unfairness is to shift the burden of taxation from the vehicle to the accessories that allow it to function. In other words, the tax must be moved back to the fuel or to the tires that record the business done by the cars shod with them. The latter seems to be the simpler expedient, and is fairer and easier to administer.

Tires cannot be used for anything but their legitimate purpose and every automobile owner must buy them. Their existence varies with the distance they cover, the rate at which they are revolved and the way in which the cars upon which they are mounted are driven. The more damage a tire does to a road, the sooner it is worn-out, and the earlier the owner has to find the money to make good the damage it inflicts. Of course, there are some parts of the country where the state of the roads is so bad that tires do not last as long as in more civilized districts, but, after all, one must pay for the luxury of living where there are good roads. Again an owner might have bad luck with one tire, but he can always change his brand. The plan would inspire the manufacture of better tires and teach drivers the virtues of carefulness and road sportsmanship.

It should not be difficult to fix such a tax. It should not vary altogether with size, but be based more on a section than on a circumference basis.—*Autocar.*

HARDWOOD LUMBER STANDARDIZATION

DURING the past fall the project of establishing new standards for the methods of grading hardwood lumber for all hardwood consuming industries was first brought to the Society's attention by the Hardwood Manufacturers' Institute in Chicago with the object of securing the cooperation of the Society as the national engineering body for the automotive industries, in carrying through this project to a successful termination. It was thought better at that time to learn more of the project before attempting to establish an organization within the Society to consider this work. It was considered, however, that probably the best method of handling it would be through the Passenger-Car Body Division of the Society's Standards Committee. Subsequently, the members of this Division were written to in this connection and the subject scheduled for discussion at the meeting of the Division in Detroit on Nov. 2, to which F. F. Murray, advisory mechanical engineer of the Hardwood Manufacturers' Institute, was invited to address the members.

He outlined the hardwood production situation and described the Institute's part in the general lumber standardization project that was largely inspired and sponsored by Secretary Hoover of the Department of Commerce, and by the National Lumber Manufacturers' Association.

Mr. Murray said:

> The Hardwood Manufacturers' Institute is national in scope, its membership comprising strictly hardwood producers and saw-mill operators whose output represents about 35 per cent of all hardwood lumber produced in the United States.
>
> Standardization of lumber sizes and cutting-up sizes for various purposes and of improved grading specifications is a problem requiring the cooperative effort of the producers and consumers, and the Hardwood Manufacturers' Institute is endeavoring to establish that cooperation with the consumers to remedy the admittedly unsatisfactory conditions that now prevail.
>
> The consumers of hardwood lumber are not alone in their complaint as to conditions within the hardwood industry as they exist to-day, as the producers suffer likewise. The cardinal purposes of the Institute are to secure adequate standardization for grading of hardwood lumber, guaranteeing the quality of shipments and meeting the consumer's requirements as fully as possible; to obstruct misrepresentation in grading, grade substitutions and the many forms of trade practice that at the present time reflect seriously on the industry.
>
> The present methods of grading hardwood are inadequate. The top grade of "Firsts" and "Seconds" is determined by specifying a permissible number of defects in the boards; for example, in a board 15 in. wide and 16 ft. long, five standard defects such as 1¼-in. knots, etc., are allowed, without regard to the location or relation of those defects. Obviously the factory man is interested in what he can get out of the boards in the cut-up sizes he wants and in the quality he gets, not merely in the number of defects in the board.
>
> The project under study is based on general factory cut-up sizes in which requirements as to quality become the units of consideration instead of the entire board being the grading unit as at present.
>
> The present methods of grading were established over a quarter of a century ago, long before the general use of the automobile and the products of many other large and specialized consumers of hardwood lumber. The present rules have not been changed in hardly any particular since then and are obsolete and inadequate in view of the tremendous developments in the hardwood consuming industries.
>
> A thorough study is necessary of the hardwod requirements of the major consuming industries, including furniture, automobiles and trucks, agricultural implements and many others, so that whatever new standards are developed will serve all of those industries to the best advantage.
>
> Lumber, unlike many other raw materials, is in a sense already manufactured by nature. The problem is to establish methods of grading and distributing so that each industry can get that part of the tree which it can use to the best advantage.
>
> This will eliminate unnecessary waste and provide a substantial saving to both consumer and producer aggregating many millions of dollars per year and the conservation of millions of feet of standing timber.
>
> The program also anticipates the building up of a satisfactory market for the lower grades of hardwood lumber, which, with prices based on supply and demand, will place the low grades in the scale with top grades and arrest the widening spread between top and intermediate or low grades of hardwoods. It is believed that therein lies the only answer to the complaint against high and increasing prices for top grades of hardwoods of which there is an actual scarcity in comparison with the lower grades.
>
> The general organization of the lumber standardization movement is centralized in the Central Committee on Lumber Standards, the organization of which was completed last October. This committee, having headquarters in the City of Washington, is an executive body representing the several branches of the lumber industry in its entirety. Ultimately the standards which the participating producer organizations develop in cooperation with the consuming industries will be grouped into a comprehensive American Lumber Standard.

Following Mr. Murray's address, discussion indicated that the body-building branch of the automotive industry is more interested in this project than any other one branch, although the motorboat, airplane, tractor and some more specialized branches of the industry are probably interested to a lesser extent. It was not felt by the members that the Society as such should endeavor to gather data and attempt to formulate standards covering the production of hardwood lumber but that the better method would be to organize a Subdivision to serve as a point of contact between the automotive industries and the Hardwood Manufacturers' Institute and to assist the Institute in an advisory capacity, providing a channel through which the Hardwood Manufacturers' Institute can obtain data if so desired. Mr. Murray stated that the view of the situation taken by the members of the Division was the same as that of the Institute and that such an organization representing the automotive industries would greatly assist in the successful completion of the program inasmuch as the automotive industry has grown to be one of the largest hardwood consumers.

George E. Goddard, chairman of the Passenger-Car Body Division, has appointed the following Subdivision to serve in this capacity.

George J. Mercer, chairman, Detroit
C. W. Avery, Ford Motor Co.
R. A. LaBarre, Towson Body Co.
F. F. Murray, Hardwood Manufacturers' Institute
Carl Simmons, Fisher Body Corporation

Inasmuch as this project will probably constitute a very important step toward the conservation of the available hardwood lumber resources of America and consequently tend toward a reduction in the cost of hardwood lumber, it is urged that all members of the Society representing hardwood consuming companies cooperate as fully as possible with the Society or the Hardwood Manufacturers' Institute when they are asked to furnish data and information regarding their hardwood lumber requirements.

Vol. XII January, 1923 No. 1

An Experimental Survey of Gasoline and Kerosene Carburetion

By C. S. KEGERREIS[1] AND G. A. YOUNG[2]

ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPHS AND CHARTS

REALIZING that great quantities of fuel are being wasted yearly by improper carburetion, the results of several investigations are assembled and considered from various viewpoints. Some pertinent data have been published previously; therefore, only a general discussion of gasoline carburetion requirements is presented in this paper. The scope of the paper is too broad to include a detailed discussion of any one subject. However, the data have been compiled to allow the graphs to explain many points that could not otherwise be touched.

The kerosene research is particularly valuable, not so much in outlining the requirements for burning kerosene as an automotive fuel exclusively, but for the carburetion requirements necessary in using our future petroleum fuels satisfactorily. Supposing that the future fuel will embody hydrocarbons boiling up to 550 deg. fahr. as a limit and containing sufficient volatile fractions to allow engine starting even in cold weather, the authors feel that kerosene will lie within the limits of efficient carburetion.

In addition to the ideal requirements, data are shown to prove the reasons for the present wastage of fuel. This is illustrated by the metering characteristics and allied data obtained from a few commercial carbureters, the data being used merely as a comparison with the ideal requirements. The change of temperature of fuel and air causes a change in the metering characteristics and the metering is not as desired. In addition, there are the temperature requirements of fuels that must be recognized, thus making the problem of carburetion truly complex.

IN the last few years our attention has been forced to center on the fuel problem more than on any other. The various aspects of this problem with respect to continued supply have been discussed thoroughly and are common knowledge to every automotive engineer. Therefore, the object of this paper is to avoid any discussion of the unmined supply of petroleum or how long it will last under the present rates of consumption, but rather to present data showing how well we can burn our present fuels and to estimate as correctly as possible some utilization requirements of our future fuels.

Formerly, satisfactory performance of our cars was easily obtained by the use of volatile fuels and, as the cost of the fuel was low, economy was of minor importance. With the advent of more than 10,500,000 cars and trucks in the United States, not including motorcycles and tractors, the situation is changed entirely. With the changing of the situation, other phases of the problem are introduced. Power is no longer paramount, being gradually forced to share its position with economy. The element of economy is now growing to be considered as essential as power. The price of gasoline is advancing steadily, yet we do not feel this to be the only reason for considering economical operation. Where power alone is concerned, too much trouble is experienced from carbon deposits, oil dilution and the cost of service and replacements due to the rich mixtures used. The United States has indeed been fortunate in securing refined fuels so easily and cheaply. However, a few years hence, when price will perhaps be the main consideration of the general public, fuel economy will be paramount.

The use of higher compressions in the throttling engines will improve economy wonderfully but, in spite of this fact, the loss due to improper mixture preparation for combustion must be eliminated to allow the best utilization. When estimated from the results of various tests, the loss of fuel in 1921 alone shows that 25 per cent of the gasoline consumed is absolutely wasted on account of improper carburetion and consequently poor combustion. The consequent $350,000,000 per year wasted would aid materially in meeting the following year's fuel bill, if conserved. Even this large sum does not indicate the total loss caused by improper fuel utilization in the United States. To the above value must be added the greater cost of service and replacements and an increased depreciation; thus, the total wastage will at least triple the above-mentioned amount. Certainly these estimates are sufficiently important for our consideration.

We must admit that the saturation point of the automotive industry is by no means attained. Viewing this fact in connection with our fuel resources, the only solution that seems practicable where the cost of the fuel is included is to refine a fuel embodying light fractions for starting and including the various hydrocarbons nearly up to the end-point of kerosene. We may not be justified in choosing a fuel of this type as the inevitable one but, when economy is considered in connection with the fuel costs, we feel the automotive industry will ultimately arrive at this conclusion.

With the increasing tendency toward heavier fuels, the automotive industry will be interested in the evaluation of some of the carburetion requirements. The Purdue Engineering Experiment Station has therefore made some researches on the subject of gasoline and kerosene carburetion. Some of the data have been presented before the Society by O. C. Berry, in his paper on Mixture Requirements of Automobile Engines,[3] and others have been published from time to time. In the present paper, additional results on gasoline are given and data on kerosene carburetion are included.

In making a study of carburetion we are interested in the ideal as well as the practical requirements. This investigation made, there will be some definite basis on which to design apparatus for the best performance with respect to the fuel in use. From another viewpoint, what metering characteristics are the various carbureters delivering to the engine? The results will show just how well our present-day devices are meeting engine requirements.

Up to the present, the researches have been confined to work on multi-cylinder engines of various types. The question of distribution in multi-cylinder engines is a

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[3] See THE JOURNAL, November, 1919, p. 364.

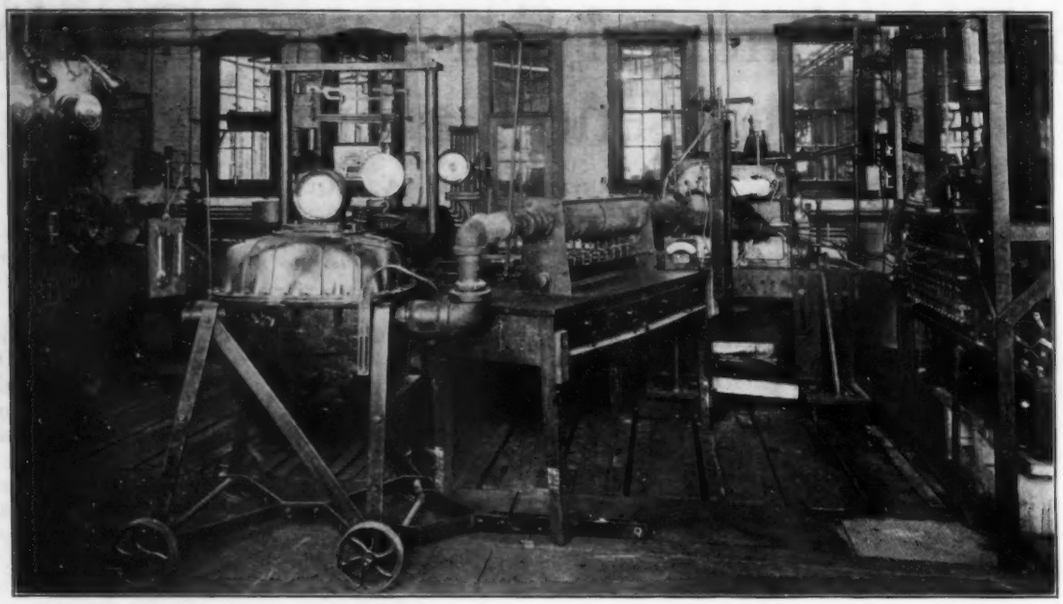

FIG. 1—DYNAMOMETER TESTING PLANT

complex problem but, when foggy or dry mixtures are induced through the average manifolds, a long stride is taken toward the solution. Since the industry is dependent on the multi rather than the single-cylinder engine, the results of this investigation are of immediate practical value.

The fuels used in the tests are the ordinary gasoline and kerosene sold at the service-stations for commercial use. The gasoline data cover the fuels used for the last 4 years, and Table 1 presents the distillation of the first fuel as well as the fuel in use now. The unsaturation content as determined by absorption in sulphuric acid is approximately 12 per cent for gasoline and 7 per cent for kerosene.

The plant on which the tests were made has been described in several publications, and its details will not be discussed at this time. Figs. 1, 2 and 3 show the entire dynamometer plant, including the fuel-weighing and electrical-control devices. All conditions were held as constant as possible, accuracy being one of the main features of testing. The data on the cooling loss probably contain the maximum error, as the heat capacity of the engine caused some lag.

TABLE 1—DISTILLATION OF FUELS USED

	Gasoline	Gasoline	Kerosene
Specific Gravity, deg. Baumé	56.5	56.5	41.2
Initial Boiling-Point, deg. fahr.	96	99	350
10 Per Cent Distilled, deg. fahr.	162	166	390
20 Per Cent Distilled, deg. fahr.	210	203	411
30 Per Cent Distilled, deg. fahr.	247	235	418
40 Per Cent Distilled, deg. fahr.	272	260	426
50 Per Cent Distilled, deg. fahr.	295	282	434
60 Per Cent Distilled, deg. fahr.	313	303	443
70 Per Cent Distilled, deg. fahr.	335	324	452
80 Per Cent Distilled, deg. fahr.	356	349	464
90 Per Cent Distilled, deg. fahr.	381	383	488
Maximum Per Cent Distilled, deg. fahr.	425	426	538

The tendency to supply less volatile fuels for the automotive industry has given rise to the question of the economy of such action. Can we obtain more car-miles per unit of crude petroleum by the use of a heavier fuel; and, if so, will the performance be satisfactory to the public? Ricardo states, in his Means of Materially Increasing Thermal Efficiency,[4] that he has made tests of about 40 fuels, and that the power and efficiency characteristics with respect to mixture strength are the same for all except alcohol. However, for satisfactory performance in an engine, the power and efficiency characteristics depend on the volatility of the fuel and hence the temperatures required for their utilization. Viewing the situation from this angle, the two fuels represented in Table 1 were tested in the same engine under the same throttle-opening and at the same speed. The pre-

[4] See *Automotive Industries*, Feb. 24, 1921, p. 456.

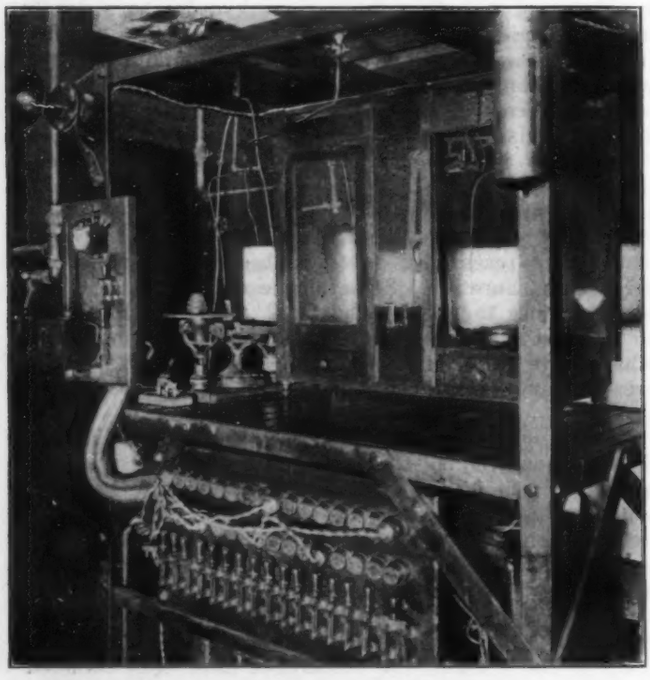

FIG. 2—FUEL-WEIGHING AND ELECTRICAL-CONTROL MECHANISM OF THE PLANT

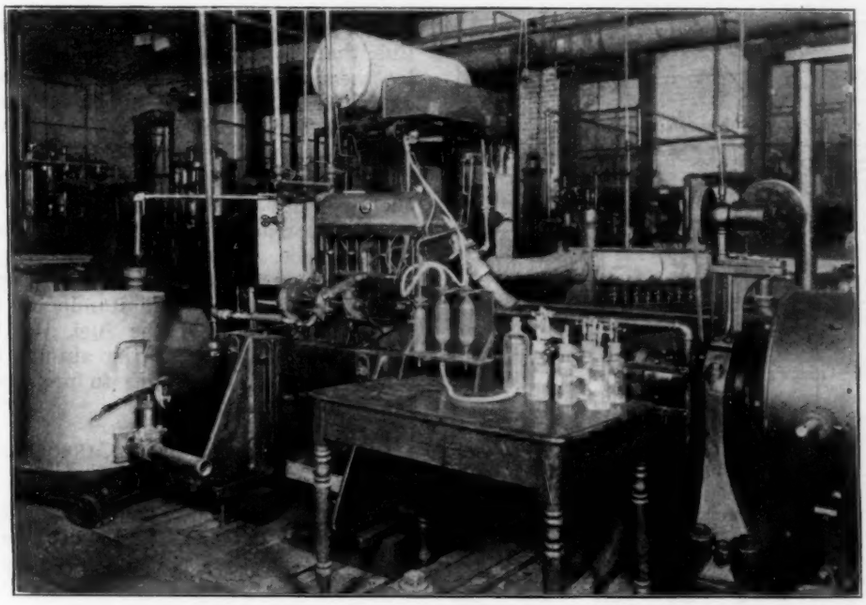

FIG. 3—THE OPPOSITE SIDE OF THE DYNAMOMETER TESTING PLANT

heated-air method was employed instead of the hot-spot in determining the proper temperature for carburetion. This method is superior for experimental research, be-

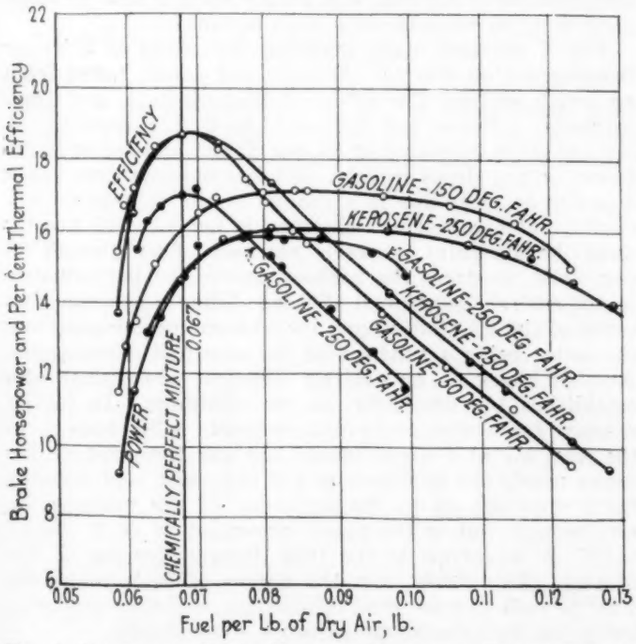

FIG. 4—POWER AND BRAKE THERMAL EFFICIENCY CHARACTERISTICS OF GASOLINE AND KEROSENE AT A CONSTANT THROTTLE-OPENING

cause of the more uniform vaporization and distribution and for the reason that the temperature can be controlled more readily. We wish to state, however, that the hot-spot is superior for practical usage in a correctly designed system of manifolding.

COMPARISON OF GASOLINE AND KEROSENE

The temperatures chosen for good performance are the minima required and allow some oil-dilution. The power and the efficiency of the two fuels are considered at this time; the temperature requirements will be discussed later in the paper. Fig. 4 presents the power and

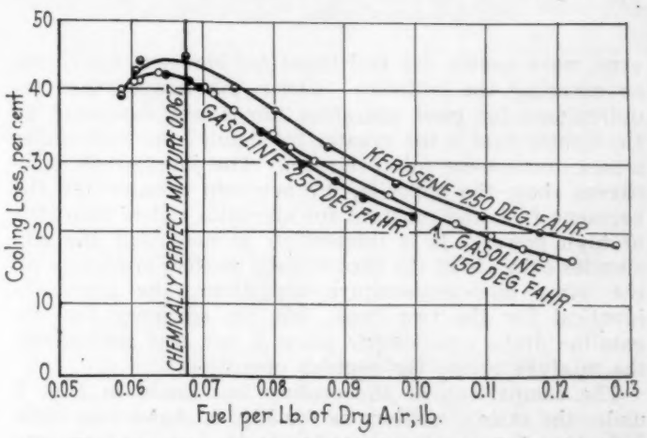
Cooling Loss, per cent
Fuel per Lb. of Dry Air, lb.
CHEMICALLY PERFECT MIXTURE 0.067
GASOLINE-250 DEG. FAHR.
KEROSENE-250 DEG. FAHR.
GASOLINE 150 DEG. FAHR.

FIG. 5—COMPARISON OF THE COOLING LOSSES FOR GASOLINE AND KEROSENE AT HALF LOAD

brake thermal-efficiency curves with respect to mixture strength. These data were taken from a four-cylinder Midwest engine of 4⅛-in. bore and 5¼-in. stroke. The gasoline-250 deg fahr. curves compare the two fuels at the same temperature; the mixtures for gasoline, how-

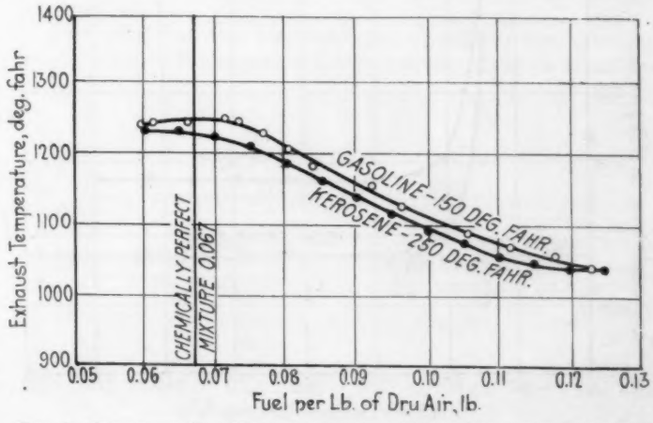
Exhaust Temperature, deg. fahr
Fuel per Lb. of Dry Air, lb.
CHEMICALLY PERFECT MIXTURE 0.067
GASOLINE-150 DEG. FAHR.
KEROSENE-250 DEG. FAHR.

FIG. 6—EXHAUST-GAS TEMPERATURES AT HALF LOAD FOR GASOLINE AND KEROSENE

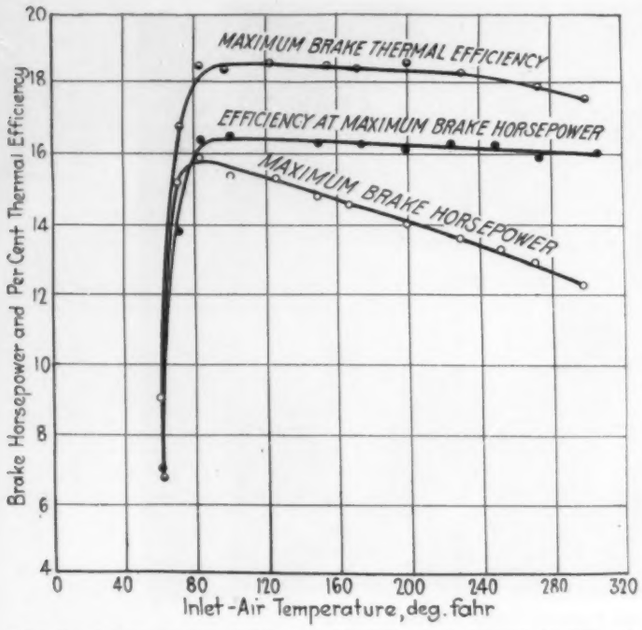

FIG. 7—EFFECT OF THE INLET-AIR TEMPERATURE ON THE POWER AND THE EFFICIENCY WHEN USING GASOLINE

ever, were nearly dry and those for kerosene fairly wet on entering the cylinder. Under the temperature requirements for good operation, the power developed by the lighter fuel is the greater by about 7 per cent under a maximum-power mixture-ratio. The thermal-efficiency curves show the opposite, 0.7 per cent greater for the kerosene fuel, and constant for all ratios richer than that of high power. It is interesting to note that the efficiencies coincide at the theoretically perfect mixture. At the same inlet-temperature conditions, the power is identical for the two fuels, but the efficiency for the gasoline drops consistently about 2 per cent throughout the mixture range for regular operation.

The comparison of the cooling loss made in Fig. 5 under the same conditions as for Fig. 4, shows very little difference for the two gasoline tests, but the kerosene series are 2.5 per cent higher. Fig. 6 gives the curves of the exhaust temperatures for the two fuels. In this instance, the average gas temperature is higher for gasoline than for kerosene. The temperatures were taken by thermocouples projecting into the exhaust-manifold at cylinder No. 1, between Nos. 2 and 3, and some distance beyond No. 4, through which region all the gases passed.

The results from these tests indicate that the heavier fuel does not develop as much power as the lighter, due to the increased temperature necessary for continued performance. The thermal efficiency, even under the higher temperatures, is as high or higher for the heavier fuel. From an economic standpoint of a continued supply and the price of the fuel, the heavier fuel is the better. It remains for the automotive engineer to develop carburetion systems to prepare the fuel properly for use in the engine.

[5] See *Automotive Industries*, Sept. 29, 1921, p. 610.
[6] See THE JOURNAL, March, 1922, p. 171.

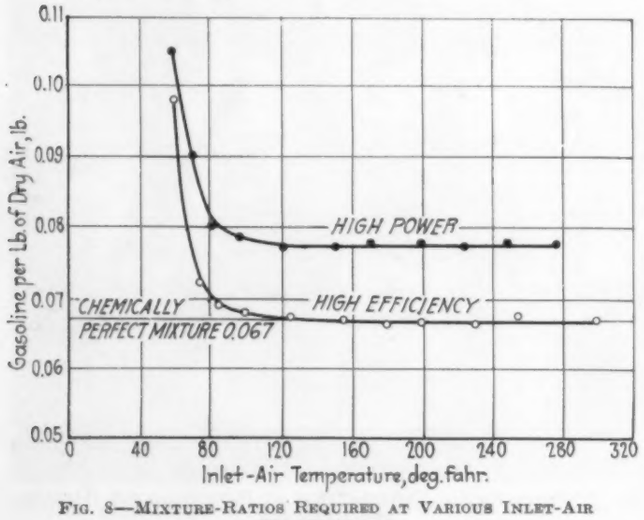

FIG. 8—MIXTURE-RATIOS REQUIRED AT VARIOUS INLET-AIR TEMPERATURES

GASOLINE CARBURETION

Since the results of some of the gasoline data compiled by the Purdue Engineering Experiment Station have been published previously, the main points only will be discussed and augmented by data in graphical form.

In the carburetion of any fuel, the four main factors linked to the metering characteristics are (*a*) the effect of the mixture temperature, (*b*) the effect of the speed, (*c*) the effect of the load, and (*d*) the effect of the compression. In all cases each fuel will require some definite temperature of air that will supply enough heat to allow good distribution and show high economy.

Fig. 7 presents data, showing the effect of inlet-air temperature on thermal efficiency and power, taken from an article entitled The Effect of Mixture-Ratio and Temperature on Power and Economy,[5] by C. S. Kegerreis. A critical point is located at 85 deg. fahr.; and below it the power output drops rapidly. Higher temperatures cause a power decrease due to a change of the volumetric efficiency, but the thermal efficiency is not affected greatly until the dry-point is nearly reached. Even though 85 deg. fahr. produces the highest power, the performance is not entirely good until 150 deg. fahr. is attained, because of the oil dilution and the uneven speed-regulation, the latter being a function of the manifold distribution. Another check on an entirely different powerplant also establishes 150 deg. fahr. as the minimum. In taking exhaust-gas temperature-measurements with respect to the inlet air at a given power, the gas temperature becomes nearly the minimum at 150 deg. fahr. and remains fairly constant up to the dry-point of the mixture, as was brought out in the paper[6] presented by O. C. Berry and C. S. Kegerreis at the 1922 Annual Meeting of the Society. This shows that the degree of heat in the air must be sufficient for good combustion, as below this critical point the exhaust temperature rises rapidly.

Taking a different viewpoint for the preceding tests, Fig. 8 presents the mixture requirements for temperature at half load.[5] The graphical results for high power and for high efficiency call for richer ratios at air temperatures of less than 125 deg. fahr.; thus, at 150 deg. fahr., the mixtures are as lean as possible for power and efficiency. Summing the results from these three viewpoints and from different engines, an inlet-air temperature of 150 deg. fahr. is the minimum that it is wise to maintain.

The limits of regular firing or inflammability are of interest. As the air temperature is decreased below 100 deg. fahr., the limits narrow abruptly,[6] as shown in Fig. 9. The leanest mixture for regular operation with wet fuel is about 150 deg. fahr. The constancy of this lean

limit under good performance temperatures signifies that the distribution is fairly uniform.

Additional information on inflammability is given in Fig. 10 for two temperatures with respect to the load. In fact, the 150 and 300-deg. fahr. series represent the low and high limits that include good operation; that is, the lower temperature represents the wettest mixture

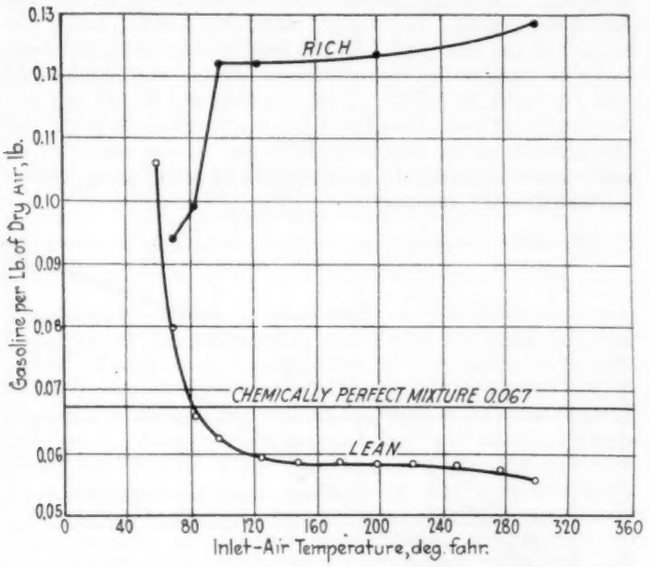

FIG. 9—THE LIMITS OF INFLAMMABILITY AT VARIOUS INLET-AIR TEMPERATURES

desirable for use, and the higher, the approximate point of dry mixtures. The dry mixtures allow better distribution by being more homogeneous, and the engine is therefore enabled to fire the poorer mixture-ratios. The effect of the load can be noted readily by comparing the

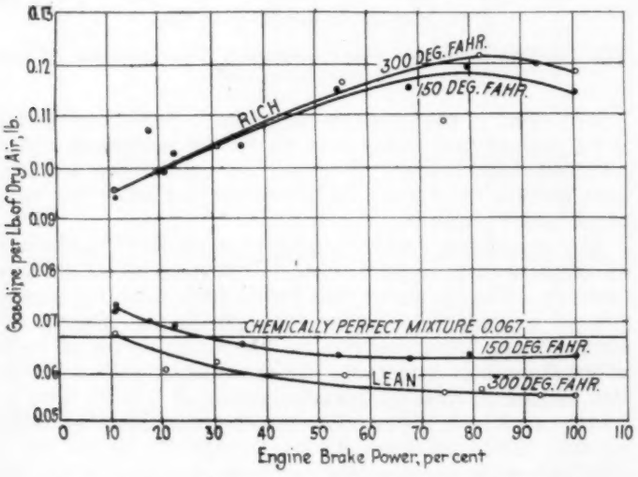

FIG. 10—THE EFFECT OF THE LOAD ON THE LIMITS OF INFLAMMABILITY

mixture-ratios required at low and high loads. At 10-per cent brake-load, a ratio of 0.072 is necessary at the lower temperature, due no doubt, to a less concentrated charge and poorer scavenging. Data from another engine are shown in Fig. 11, illustrating the effect of the speed at the different loads. A point worthy of mention in connection with regular firing is the rich mixture required for about 10 per cent brake-load at the lean limit. In fact, the engine developed nearly the maximum results

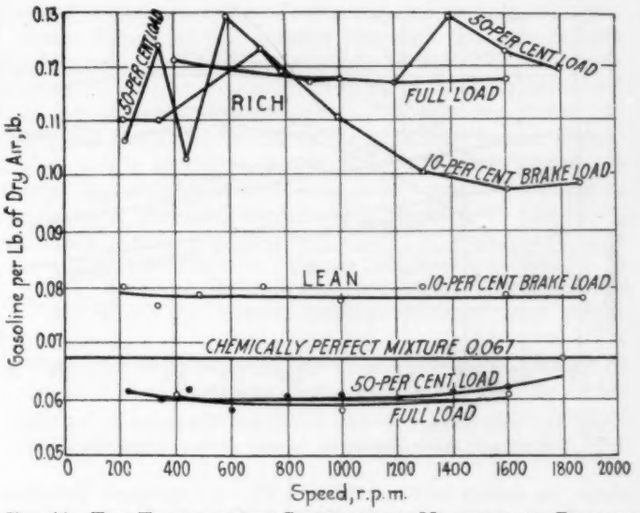

FIG. 11—THE EFFECT OF THE SPEED ON THE MIXTURES FOR REGULAR INFLAMMABILITY

when using this ratio, regardless of occasional misfiring. Manifolding is a factor in this case, as well as the mechanical condition of the engine; for example, valve-stem

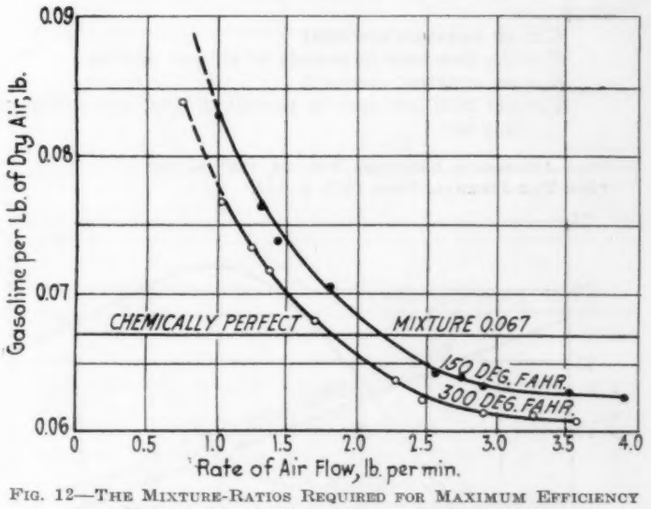

FIG. 12—THE MIXTURE-RATIOS REQUIRED FOR MAXIMUM EFFICIENCY AT VARIOUS LOADS, EXPRESSED IN AIR-FLOW RATE

wear, gasket tightness and the like. The higher loads, at any speed tested, show nearly the same ratios for all engines studied.

The effect of the load and the speed on the mixture requirements for gasoline follow certain definite relation-

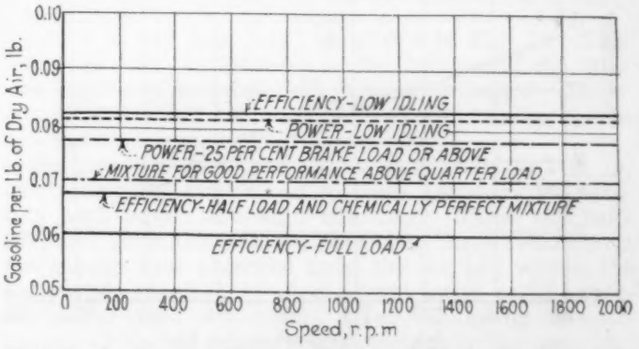
EFFICIENCY-LOW IDLING
POWER-LOW IDLING
POWER-25 PER CENT BRAKE LOAD OR ABOVE
MIXTURE FOR GOOD PERFORMANCE ABOVE QUARTER LOAD
EFFICIENCY-HALF LOAD AND CHEMICALLY PERFECT MIXTURE
EFFICIENCY-FULL LOAD
Gasoline per Lb. of Dry Air, lb.
Speed, r.p.m.

FIG. 13—THE MIXTURE-RATIOS REQUIRED FOR GASOLINE AT VARIOUS LOADS AND SPEEDS

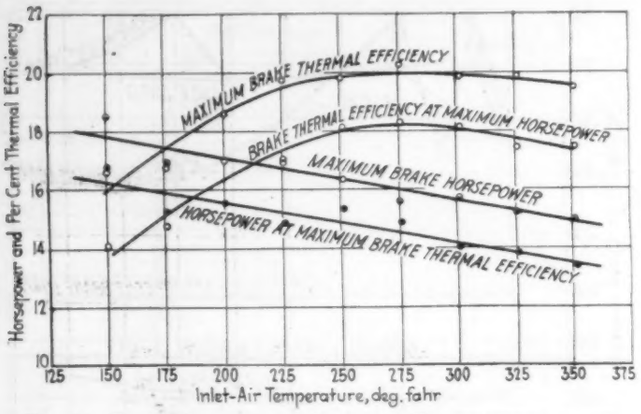
Horsepower and Per Cent Thermal Efficiency
MAXIMUM BRAKE THERMAL EFFICIENCY
BRAKE THERMAL EFFICIENCY AT MAXIMUM HORSEPOWER
MAXIMUM BRAKE HORSEPOWER
HORSEPOWER AT MAXIMUM BRAKE THERMAL EFFICIENCY
Inlet-Air Temperature, deg. fahr

FIG. 14—THE EFFECT OF THE INLET-AIR TEMPERATURE ON THE ENGINE CHARACTERISTICS WHEN USING KEROSENE

ships, as shown in Figs. 12 and 13. An example pointing out the requirements of efficiency for any change of the load is found in Fig. 12, where the flow of air represents the load at a constant speed. The curve representing mixture-ratio versus flow-rate is of a hyperbolic form and can be represented by the general equation

$$M = C + K/F$$

where

C = an equation constant
F = the flow-rate in pounds of air per minute
K = an equation constant
M = the mixture-ratio in pounds of fuel per pound of dry air

[7] See *Automotive Industries*, Feb. 24, 1921, p. 460.
[8] See THE JOURNAL, June, 1921, p. 553.

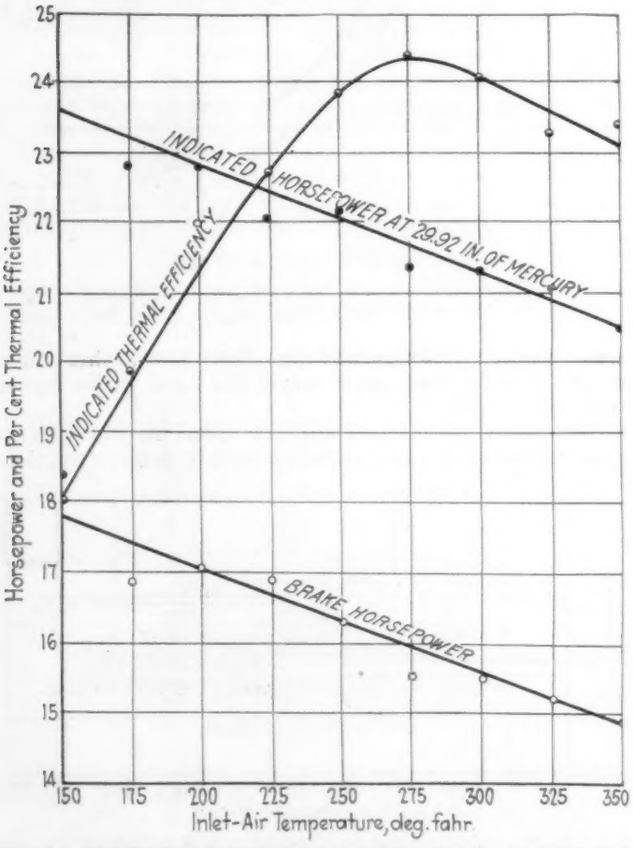
Horsepower and Per Cent Thermal Efficiency
INDICATED HORSEPOWER AT 29.92 IN. OF MERCURY
INDICATED THERMAL EFFICIENCY
BRAKE HORSEPOWER
Inlet-Air Temperature, deg. fahr.

FIG. 15—THE EFFECT OF THE INLET-AIR TEMPERATURE ON THE POWER AND THE EFFICIENCY IN TERMS OF THE INDICATED RESULTS

For the test series at 150 deg. fahr., the equation assumes the form

$$M = 0.0545 + 0.0283/F$$

and, at 300 deg. fahr. or for the dry mixtures,

$$M = 0.0553 + 0.0210/F$$

The effects of the load and the speed are shown in Fig. 13. The high power-ratio for variation of the load changes slightly, the richer mixture, 0.0815, being required at the low load. As the load is increased, the ratio becomes constant at approximately 25-per cent brake-load and its value is 0.0775 lb. of gasoline per lb. of dry air. Formerly, it was thought a constant ratio sufficed; but the change in the mixture-ratio, due to the variation of mechanical accuracy in engines and to scavenging, makes a small change necessary.

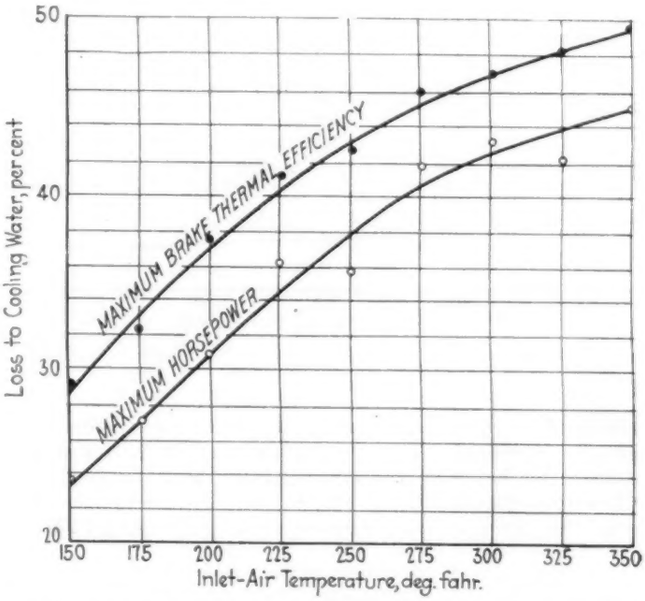

FIG. 16—THE EFFECT OF THE CARBURETION TEMPERATURES ON THE ENGINE COOLING LOSS

The effect of the speed on the mixture-ratio is nil under good temperature conditions up to the maximum speed attained, 2000 r.p.m. With correct carbureter and engine design, the higher speeds should not affect the constant ratios required.

The remaining factor, compression, with its influence on mixture-ratio requirements, is not determined so conclusively. Ricardo states that for all fuels, with the exception of alcohol, the mixture strength is not affected by compression.[7] Sparrow relates, "A recent series of tests at the Bureau of Standards has shown that, with an aviation engine having compression-ratios of 5 to 3, 6 to 3, 7 to 3 and 8 to 3, the maximum power was obtained with air-fuel ratios of from 13 to 1 (0.0770) to 14 to 1 (0.0714) with air densities of from 0.025 to 0.075 lb. per cu. ft. This was, however, with aviation gasoline, a more volatile product than commercial fuel."[8] Fragments of data from other sources show a tendency toward the poorer mixture-ratios at the higher compressions. It is reasonable to maintain that the more concentrated the charge is, the less will be the fuel necessary in the mixture for the maximum power development.

KEROSENE INVESTIGATIONS

In dealing with kerosene as a fuel, the same factors must be evaluated in the same manner as with gasoline. The temperatures for satisfactory operation must first be

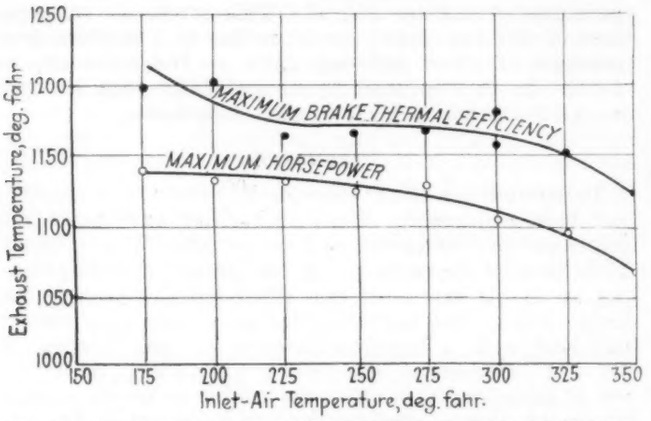

FIG. 17—THE REGISTER OF THE EXHAUST TEMPERATURES FOR VARIOUS INTAKE TEMPERATURES

determined. Since the volatility of kerosene is lower than that of gasoline, the temperatures required for good power and efficiency development will be relatively higher. With the same throttle-opening Fig. 14 presents the effect of the air temperature on the power and the thermal efficiency. Kerosene at one-half load will develop higher power at 150 deg. fahr. than at any higher temperature, even though the engine operation is very erratic and the oil dilution excessive. The thermal efficiency proves the poor operation at such low temperatures. Maximum brake thermal-efficiency occurs at 275 deg. fahr.; and the indicated thermal-efficiency occurs at the same point as shown in Fig. 15. Considering power and economy, temperatures lower than 250 deg. fahr. should not be used and, in view of the excessive oil-dilution, this is only too true.

Some details of the cooling loss and the exhaust-gas temperatures are plotted in Figs. 16 and 17. The percentage of cooling loss with respect to inlet-air temperature mounts as a straight line up to 250 deg. fahr.; at this point the slope of the curve begins to break, resulting in lower heat-loss per unit of air-temperature increase. The inlet and outlet jacket-water was maintained at a constant temperature throughout the tests, the outlet-water temperature being 150 deg. fahr. The register of the average exhaust-temperatures, shown in Fig. 17, is given for the mixture-ratios corresponding to the maximum brake thermal-efficiency and brake horsepower.

The limit of inflammability is another point worthy of consideration in proving more completely the optimum temperature for engine performance. The lean limit of regular firing breaks into a straight line at an air temperature of 250 deg. fahr., as shown in Fig. 18. This being the case, an increase in the dryness of the mixture would not materially aid in the distribution. Below this critical point, the mixture limit for regular operation increases rapidly in richness.

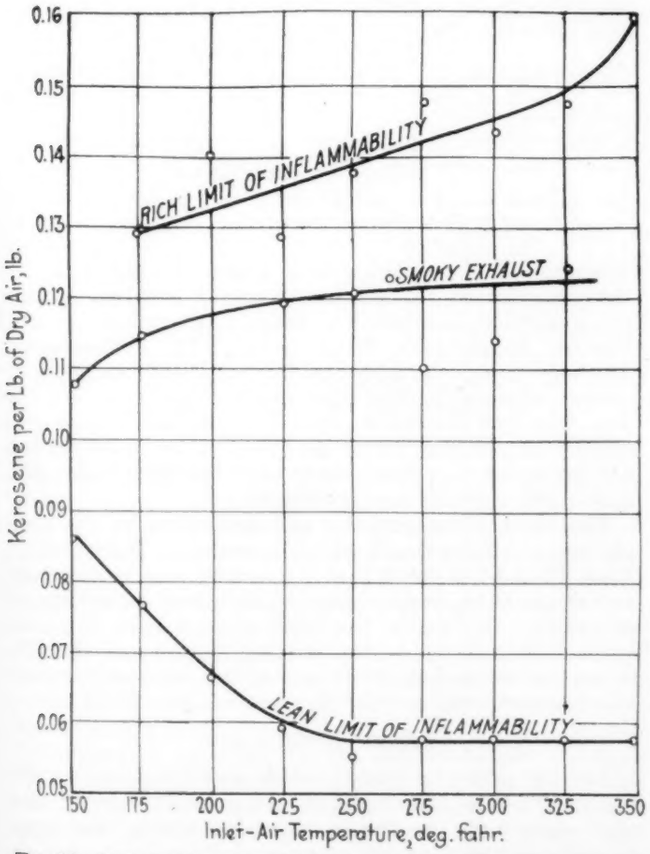
RICH LIMIT OF INFLAMMABILITY
SMOKY EXHAUST
LEAN LIMIT OF INFLAMMABILITY
Kerosene per Lb. of Dry Air, lb.
Inlet-Air Temperature, deg. fahr.

FIG. 18—INFLAMMABILITY OF KEROSENE-AIR MIXTURES WITH RESPECT TO THE INLET TEMPERATURE

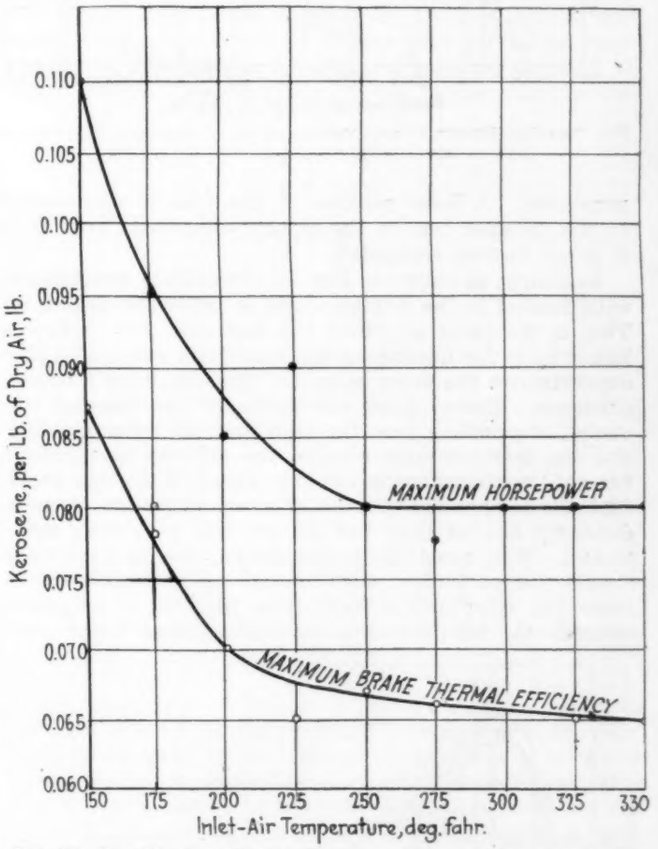

FIG. 19—THE MIXTURE-RATIOS NECESSARY FOR THE MAXIMUM POWER AND EFFICIENCY DEVELOPMENT USING 41.5 BAUME KEROSENE

The point in question is shown graphically in Fig. 18, illustrating the richness of mixture necessary to maintain high power and high efficiency. Under the half-load and lower-temperature condition, there evidently is not enough heat absorbed from the air and within the combustion-chamber to allow a sufficiently rapid vaporization for quick combustion. A small change in the heat-content of the air causes a large change in the completeness of combustion. The cooling-loss curves in Fig. 15 present very nicely the low loss occurring at low tem-

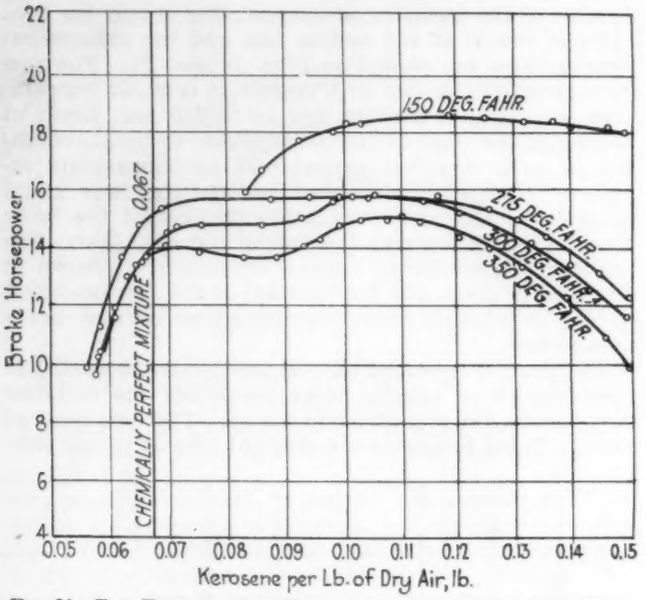

FIG. 20—THE EFFECT OF THE TEMPERATURE ON THE RELATIVE INTENSITY OF DETONATION FROM A POWER STANDPOINT

peratures. A large portion of the fuel is not burned in the cylinder but in the exhaust-manifold; even then it is not burned completely.

Evidently, as shown in Fig. 19, the critical temperature with respect to the mixture-ratio is again 250 deg. fahr. This is the point at which the fuel does not require a leaner ratio for developing the maximum power and also approximates the same point for the maximum thermal-efficiency. Hence, from the values of the thermal efficiency, the cooling loss, the lean limit of inflammability and the mixture-ratio results, the critical temperature for good performance is 250 deg. fahr. With this heavy fuel, the temperature for satisfactory operation is more definitely located than for the gasoline previously mentioned. For practical performance, the temperatures should not go higher, even though more trouble exists from the oil dilution; but, if a method is employed whereby the oil dilution is minimized, then lower temperatures should be utilized. This intake-air temperature of 250 deg. fahr., corresponding to a mixture temperature of about 205 deg. fahr. on the wet bulb, no doubt can be decreased by exceptionally good hot-spot design if the hot-spot allows good distribution.

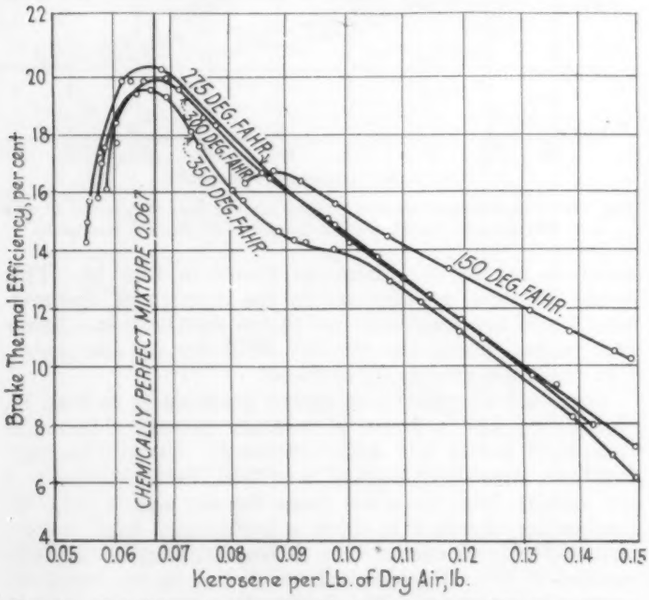

FIG. 21—THE EFFECT OF THE TEMPERATURE ON THE RELATIVE INTENSITY OF DETONATION FROM THE STANDPOINT OF EFFICIENCY

DETONATION

Detonation is a factor entering into these tests that has not been mentioned. Some authorities have made the statement that temperature does not affect the intensity of detonation appreciably. A few graphs are enlightening as to the extent of this effect with respect to the temperature. The tests reported previously were run at half-load with a compression-ratio at open-throttle of 4.3 to 1. Detonation was present at 275 deg. fahr., but not of an intensity to affect the power, as shown in Fig. 20, or the thermal efficiency, as is brought out in Fig. 21. An increase of 25 deg. fahr. gave the result shown on the 300-deg. fahr. curves. There is a loss of at least 1 b. hp., as well as a slight decrease in the thermal efficiency. The effect of intense detonation is noted on the 350-deg. fahr. curves by a loss of 11.0 per cent in power and 1.5 per cent in thermal efficiency; and the mixture

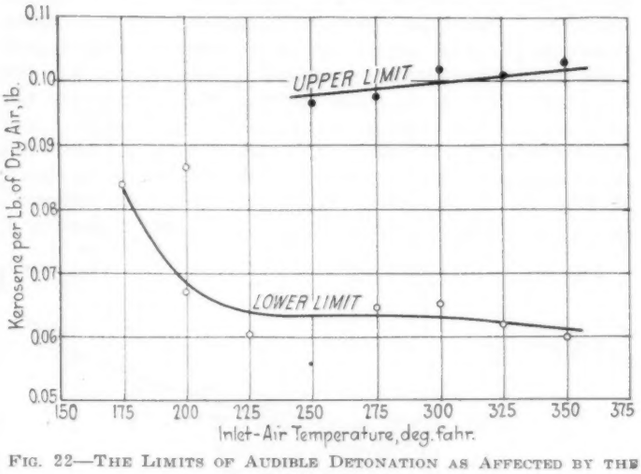

FIG. 22—THE LIMITS OF AUDIBLE DETONATION AS AFFECTED BY THE INTAKE TEMPERATURE

strength can be varied widely before this effect is entirely eliminated. When the detonation is severe enough to be audible, the mixture range with respect to the inlet-air temperature shown in Fig. 22 is interesting. The lean limit is the more regular, but is only an approximation because the listening method was used. At 150 deg. fahr., no detonation could be detected; but it was present at 175 deg. fahr. At 225 deg. fahr., detonation existed at about as lean a ratio as at the higher temperatures; but it was of less intensity.

The effect of temperature and detonation on the cooling loss and the exhaust-gas temperatures, illustrated in Figs. 23 and 24, shows that the cooling loss is increased and the exhaust temperature is decreased under severe detonation. Evidently, the combustion is more complete before the exhaust occurs, yet the power and the efficiency are decreased on account of the excessive detonation pressures which result in an increased cooling loss.

EFFECT OF THE LOAD

Another series of investigations was designed to show the effect of the load on the mixture requirements. The tests were made on the same engine and at the same speed as in the previous investigation, using the lowest air-temperature to give good performance. The com-

pression was lowered to 85 lb. per sq. in. absolute, which reduced the intensity of the detonation at the higher loads. At full loads, the detonation remained excessive and diethyl selenide was used to suppress it. A register of the inlet-air and the mixture temperatures is found in Fig. 25. Since the mixture was not entirely dry, wet-bulb temperature-readings are given; moreover, the change with the increased load shows a decrease in the mixture temperature.

Results reported in terms of brake horsepower are perfectly correct for application to some few engines; but,

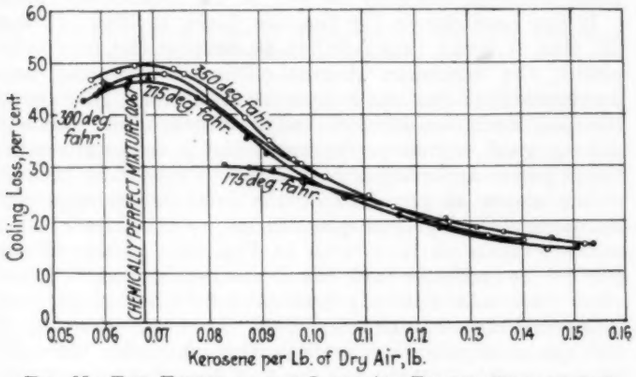

FIG. 23—THE EFFECT OF THE INLET-AIR TEMPERATURE ON THE COOLING LOSS

as very few powerplants consume the same percentage of indicated horsepower as frictional loss, the results must be reported in terms of indicated horsepower to be universally applicable. In these investigations, orifices were used for throttling and the friction was determined, under as near the operating conditions as possible, by the use of the dynamometer as a motor. The friction con-

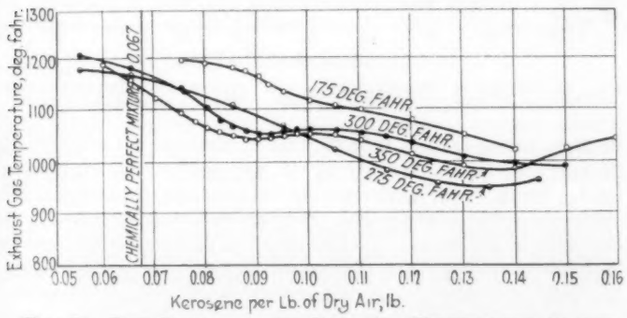
Exhaust Gas Temperature, deg. fahr.
Kerosene per Lb. of Dry Air, lb.
175 DEG. FAHR.
300 DEG. FAHR.
350 DEG. FAHR.
275 DEG. FAHR.
CHEMICALLY PERFECT MIXTURE 0.067

FIG. 24—THE EFFECT OF THE INLET-AIR TEMPERATURE ON THE TEMPERATURE OF THE EXHAUST GAS

sumed or the change of the pumping loss was found with respect to the indicated power and plotted in Fig. 26. The values of the manifold-depressions are included to help illustrate the increase of the friction at the lower loads. This engine requires 56 per cent more power for internal losses at idling than at full load; thus, the throttling increases the rate of fuel-consumption at the low loads.

From the foregoing tests, it is now possible to delineate the mixture requirements of kerosene with respect to the load and in terms of the indicated horsepower. Knowing the requirements for high power and thermal efficiency and the point of zero brake-load, the trend of the mixture-ratios necessary is shown in Fig. 27. Absolute idling, when the maximum power is desired, requires a ratio of 0.0975 lb. of kerosene per lb. of dry air. The poorest ratio required, 0.0800, is not reached until about 48 per cent of the indicated load. At 75-per cent load

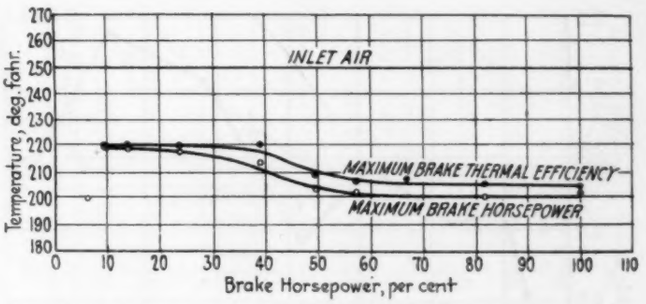

FIG. 25—THE REGISTER OF THE MIXTURE TEMPERATURES AT VARIOUS LOADS

and above it, the mixture enriches slightly, but this is partly due to the slow increase of the power with respect to the increase of the mixture strength. For example, a ratio of 0.0800 develops 33.5 i.hp. and a ratio of 0.0960, 33.9 i. hp.; this is an increase of 1.2 per cent in the power through a range of 19 per cent of the mixture strength. The high-power point is taken as a mean at about 0.0850. An increase of the temperature would

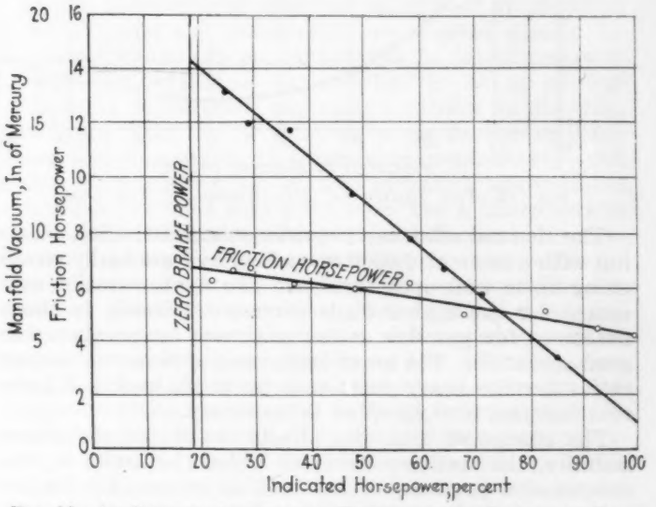

FIG. 26—AN ILLUSTRATION OF THE EFFECT OF THE CHANGE OF LOAD ON THE ENGINE PUMPING LOSS

cause, no doubt, a change of this maximum-power to 0.0800. The theory that the fuel is vaporized in the combustion-chamber by the heat of compression is no doubt true; but, when depending on the compression temperatures, the difficulty is that these temperatures are not of sufficient duration to allow vaporization and a thorough mixing prior to combustion.

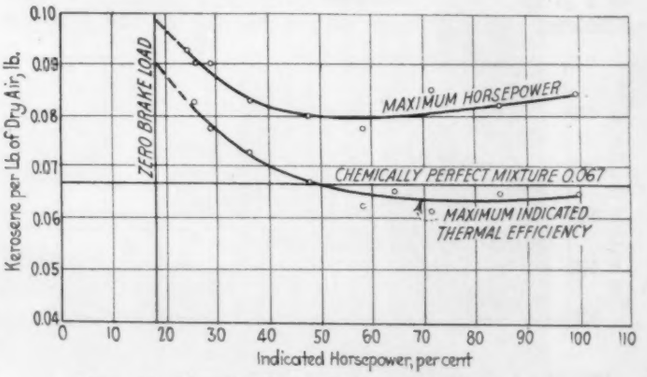
Kerosene per Lb. of Dry Air, lb.
ZERO BRAKE LOAD
MAXIMUM HORSEPOWER
CHEMICALLY PERFECT MIXTURE 0.067
MAXIMUM INDICATED THERMAL EFFICIENCY
Indicated Horsepower, per cent

FIG. 27—MIXTURE-RATIO REQUIREMENTS FOR LOAD

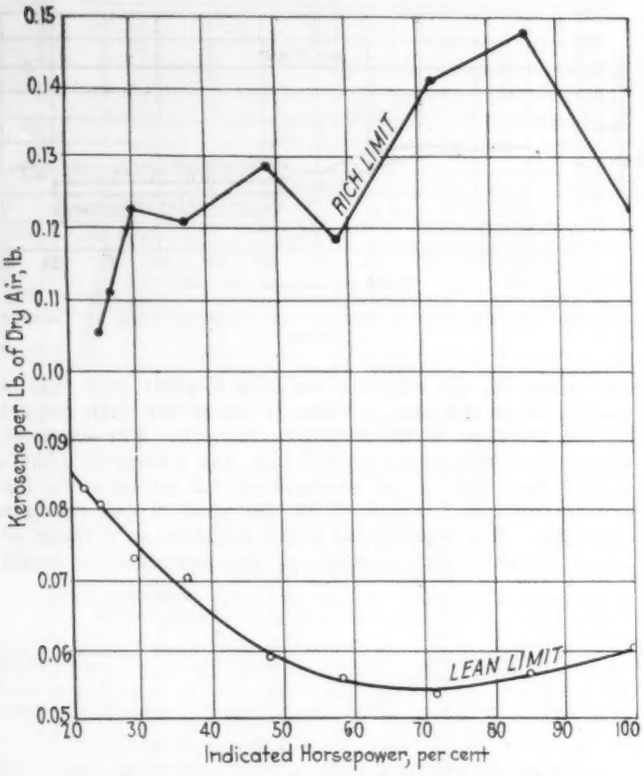

FIG. 28—THE LIMITS OF INFLAMMABILITY FOR LOAD

The thermal efficiency requirements are similar, starting with a ratio of 0.0900 at no load and gradually weakening up to wide-open throttle. The mixture-ratios necessary for the higher loads correspond closely to those necessary for gasoline at the minimum temperature for good operation. The lower loads require somewhat richer ratios for the heavy fuel; that is, at no load, a 9.1-per cent increase over gasoline is necessary.

The change of load also affects the limits of inflammability, the tendency being to widen the limits as the compression pressure or the load is increased. Distribution and mixing at the rich ratios cause an irregular line, as shown in Fig. 28, but the lean limit, which is the more valuable in all cases, is fairly uniform. In no instance does the missing or back-firing occur with mixtures as rich as those required for high efficiency or high power.

FUEL-AIR MIXTURE REQUREMENTS FOR ENGINES

Some authorities prefer to analyze the mixture requirements for engines with respect to the development of a given power. Under throttled conditions, the thermal efficiency will increase with a leaning of the mixture-

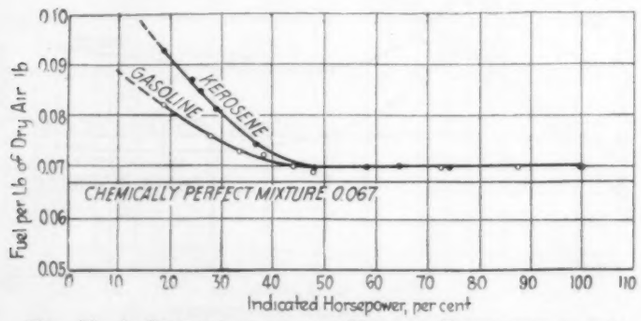

FIG. 29—A COMPARISON OF THE MIXTURE REQUIREMENTS FOR GASOLINE AND KEROSENE WITH RESPECT TO THE LOADS

ratio and will continue to do so until the air-cycle efficiency is approached as a limit. It happens that the limit of inflammability for most engines is 0.0600 or slightly leaner. In other words, to develop a given power as the mixture-strength is weakened beyond the high-power point, the larger throttle-opening necessary causes a higher compression and, hence, a higher thermal-efficiency. Thus, from the standpoint of the engine, where the load is the prime requisite, the analysis under discussion cannot be used. Then, too, at wide-open throttle the analysis must revert from a given power to a constant compression with respect to the mixture-strength.

It has been shown for the two fuels, in Figs. 13 and 27, that it is an impossibility to develop the maximum power, the maximum thermal-efficiency and good performance with the same mixture-strength. This being the case, a mixture-strength must be determined that will deliver good engine performance and a compromise between power and efficiency; and this is especially true at points above 48 per cent of the indicated horsepower. Kerosene, with proper preparation, will operate with mixture-ratios as illustrated in Fig. 29. Below 48 per cent of the indicated load, the ratios required grow somewhat rich. At 18.3 per cent of the indicated or zero brake-load, the mixture is 0.0925 lb. of fuel per lb. of dry air or slightly richer than the maximum thermal-efficiency ratio. The ratio for 56.5-deg. Baumé gasoline

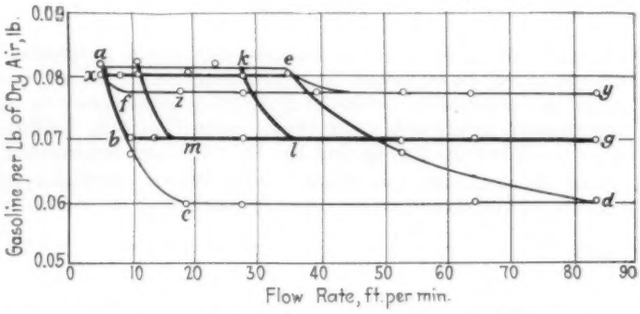

FIG. 30—MIXTURE REQUIREMENTS FOR POWER AND EFFICIENCY AT VARIOUS SPEEDS AND LOADS WITH 56.5 BAUME GASOLINE

at zero brake-load is 0.0825, and it decreases to 0.0700 lb. of fuel per lb. of dry air at 42 per cent of the indicated load. Both fuels will develop nearly the maximum efficiency until a mixture-strengeh of 0.0700 is reached. As the load is increased, it will be noted that 95 per cent of the power limits the mixture to 0.0700 for the two fuels; and this is the power sacrifice necessary to secure a higher efficiency and yet not impair the engine operation.

Knowing the effect of the speed and the load on the engine mixture requirements and operating under satisfactory temperature conditions, a graph can be drawn with respect to flow-rate that is of special interest to carburetion engineers. In Fig. 30, the mixture-ratios for gasoline are plotted against flow-rate, which can be expressed in either weight or volume per unit of time. The limits for maximum efficiency at a low speed are shown by the curve *a b c* in Fig. 30, and at some higher speed by the curve *e d*. The limit for power at the low speed is the curve *x f z*, and at the higher the curve *e y*. From the data in Fig. 29, the good-performance mixture follows very closely the maximum-efficiency ratios until about 42 per cent of the indicated load is reached; it then maintains a constant mixture; hence we have the ratios *a b m*. Similarly, for a higher speed, the curve *e g* represents the metering characteristics. Separating the metering characteristics from the power and efficiency graph, Fig. 31 presents the ratios that a carbureter must deliver to allow the best engine performance. The only manner in which

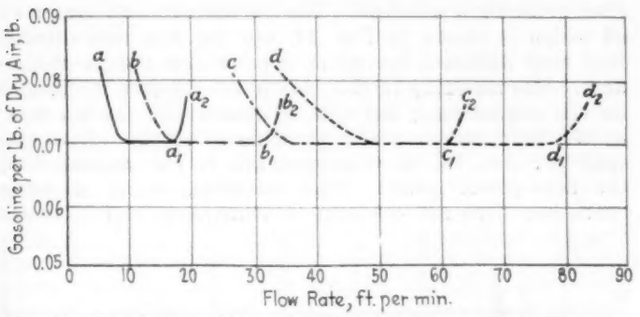

FIG. 31—THE ENGINE MIXTURE REQUIREMENTS FOR GOOD PERFORMANCE USING 56.5 BAUME GASOLINE AS FUEL

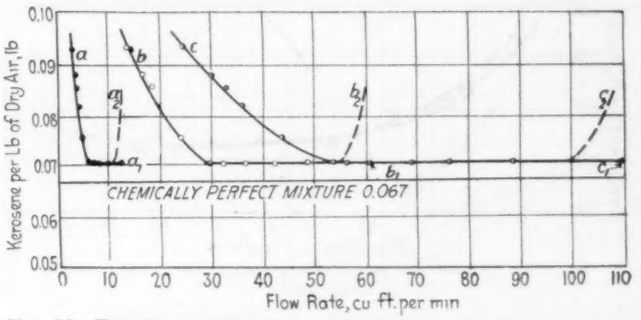

FIG. 32—THE ENGINE MIXTURE REQUIREMENTS FOR GOOD PERFORMANCE USING 41.5 BAUME KEROSENE AS FUEL

the characteristics are changed is by causing the mixture to return to the maximum power-value, a 5-per cent power change, at open-throttle positions, as this is the main interest at this condition. The curve $a\ a_1\ a_2$ illustrates the requirements at low speed; any one of the other curves, such as $d\ d_1\ d_2$, shows the requirements at the higher speeds.

The metering characteristics for kerosene are derived similarly to those for gasoline, but call for a larger variation of the ratio with respect to the load, as shown in Fig. 32. Absolute no-load, or zero brake-load, requires 0.0975 lb. of fuel per lb. of dry air; this decreases to 0.0700 at 48-per cent indicated load and just before the open-throttle and more fuel is demanded until a mixture of 0.0850 is reached at the open position. The metering characteristics outlined for kerosene represent the results from only one engine. The gasoline data are more complete, as several engines of different types have proved the same thing. Even though the data from kerosene are meager with respect to other engines, the results will blaze the way and provide the limit to which the metering characteristics for engine operation will tend to go with fuels of decreased volatility.

CARBURETER REQUIREMENTS

When economy is the chief interest in operating throttling internal-combustion engines, the ideal toward which we must strive in practice is somewhat different from that previously explained. The requirements mentioned were from the standpoint of the engine, and they do satisfy its power and efficiency requirements. When the engine is placed in a car and the car requirements for satisfactory performance with maximum economy are outlined, the carbureter is changed of necessity. The first requisite is easy starting; the second, good idling; the third, instantaneous acceleration; and the fourth, economical constant-speed and load operation.

Proper temperatures must be maintained at all costs, as distribution and combustion are affected greatly by the temperature. Proper atomization is not of necessity a requirement; but it is essential in using present methods of carburetion and assists greatly in starting. The idling must be at mixture-ratios developing high power and comparable to the engine requirements. All points of constant-speed or load operation above idling and under throttled conditions must use a mixture-ratio that is just rich enough for regular inflammability. As the ratio is weakened, the efficiency increases with the air cycle as a limit; so, the limits of inflammability are those of practical operation. For constant speed, the throttle position does not matter and the analysis with respect to a given power applies. The difficulty arises when a change of the speed and the load is necessary and, unless the ratios are returned to the high-power strength, poor operation will result. The requirement for acceleration is then to change the mixture-strength to 0.0775 for gasoline, or 0.0850 lb. of fuel per lb. of dry air for kerosene instantaneously and temporarily; then to re-

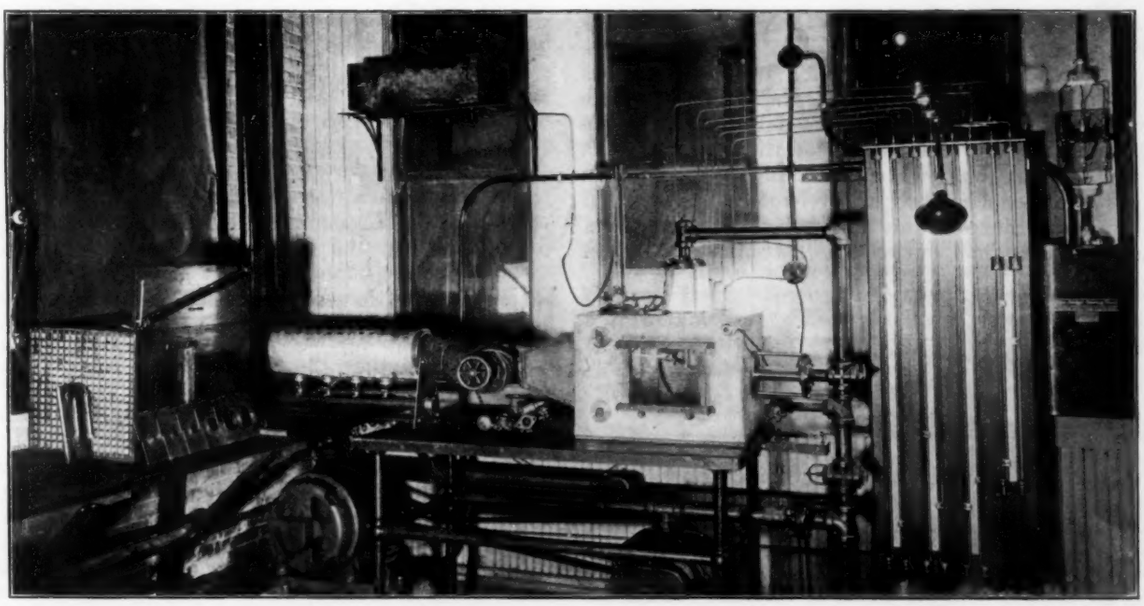

FIG. 33—THE PLANT FOR TESTING COMPLETE OR COMPONENT PARTS OF CARBURETERS

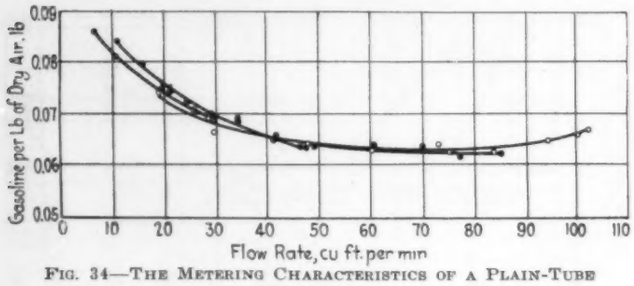

FIG. 34—THE METERING CHARACTERISTICS OF A PLAIN-TUBE CARBURETER

turn to the original lean-ratio at constant conditions. When open-throttle is attained, maximum power is the main consideration; this again requires that the mixture ratio be enriched to 0.0775 for gasoline and 0.0850 for kerosene and permanently held there for the maximum load. It will be noted from this discussion that the car requirements for constant speed differ from

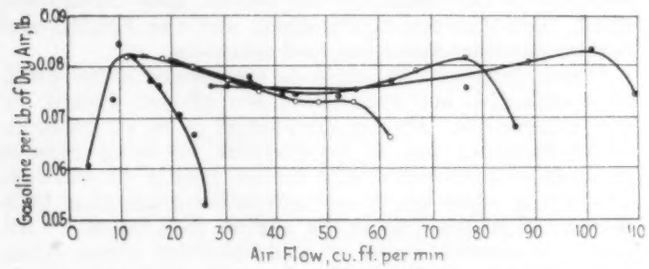

FIG. 35—THE METERING CHARACTERISTICS OF A COMMERCIAL CARBURETER

those of the engine only in that the mixtures are leaner between idling and full throttle.

COMMERCIAL METERING CHARACTERISTICS

How nearly the metering characteristics of various commercial carbureters fulfill the ideal, can be noted from the curves that present the results of investigations made with the new carbureter-testing plant, at the Purdue University Engineering Experiment Station, Fig. 33, which is similar in some respects to that at the Bureau of Standards.

Two methods were employed in these tests. Instead of making the usual mixture-ratio versus flow-rate curve from idling to the maximum capacity, the flow-rate was broken into at least four sections representing the same number of engine speeds; then, the carbureter throttle was operated only for the change of the load. In this manner many irregularities of metering can be shown that the conventional method or level-road operation will not discovered. Since the ideal carbureter requirements are known, this method compels the carbureter to recognize more than air-flow. The carbureter, the metering of which is shown in Fig. 34, has the flow-rate divided into four different speeds to present the change-of-load data. The metering in this case is not affected materially by the engine load, but almost entirely by the air-flow, as the four curves nearly coincide at idling. At wide-open throttle, the mixture-strength is not increased to the high-power point. This metering, while allowing operation, does not conform to economical and satisfactory car-performance. In the case of the carbureter presented in Fig. 35, the metering is entirely different from the ideal. At the two medium speeds the idling is good; but it is poor at the low speed. At wide-open throttle, the mixture leans rather than enriches and, at the low speed, it is so lean that the engine probably would not operate. At throttled conditions the leaning is not sufficient, thereby wasting fuel. The results of a third type of carbureter shown in Fig. 36 do not more nearly approach the ideal. In the discussion of this third type, it is assumed that the adjustment will be transferred nearer to the ideal mixture-strength. The idling at the extreme low speeds is good; but, at higher speeds, it is uncertain on account of lean ratios. At other throttled loads, the ratios are strengthened instead of weakened, and in all the higher speeds the maximum power would not be developed owing to the weakened ratios. The ratios between 15 and 30 cu. ft. per min. are erratic. Check-tests made were perfectly good at the higher flow-rates, but any mixture-ratio might be expected between the limits mentioned. Judging from the few examples

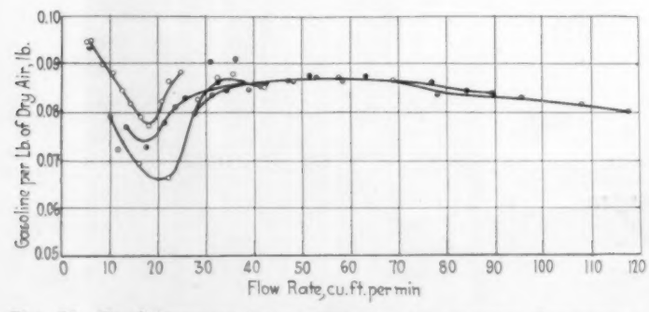

FIG. 36—THE METERING CHARACTERISTICS OF ANOTHER COMMERCIAL CARBURETER

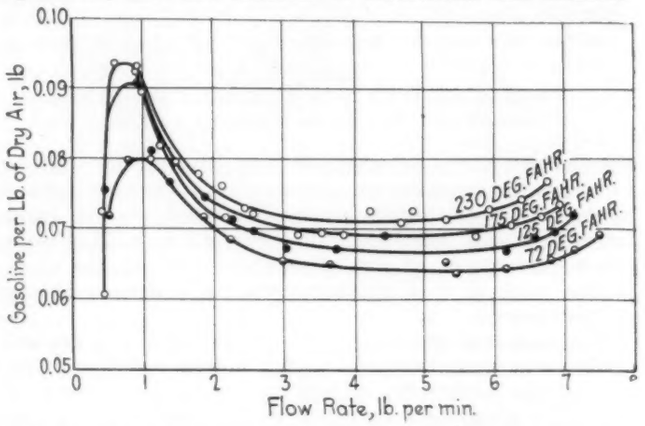

FIG. 37—THE EFFECT OF THE TEMPERATURE ON THE METERING OF A COMMERCIAL CARBURETER

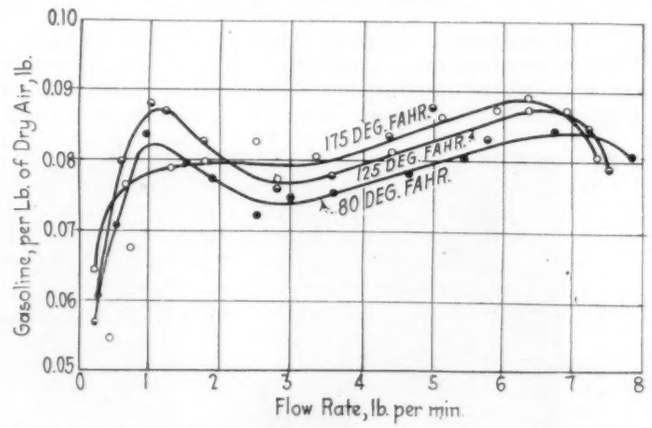

FIG. 38—THE EFFECT OF THE TEMPERATURE ON THE METERING OF ANOTHER COMMERCIAL CARBURETER

shown at constant speeds and loads, the metering is by no means ideal.

The change of the temperature, of both the fuel and the air, affects the metering. The influence of the fuel temperature can be controlled by the design of nozzles, jets and passageways, but the effect of the air-temperature requires a more complicated design. The carbureter in the two cases shown, in Figs. 37 and 38, illustrates the change of mixture-ratio at the same flow-rates for a few temperatures. In this instance the test approximates the metering characteristics delivered when operating on a level road; that is, change of the speed and the load combined. The carbureter, the characteristics of which are illustrated in Fig. 37, enriches about 8 per cent per 100-deg. Fahr. increase. The other device at a slightly richer mixture shows 7 per cent per 100-deg. Fahr. change. The temperatures studied are from 75 to 225 deg. Fahr.; they cover the ordinary range with the exception of the low temperatures after starting. In designing thermostatic controls, other temperatures must be considered in addition to those of the air and the fuel.

The question of acceleration is of prime importance at any time. If an instrument is designed to fulfill the carbureter requirements, the acceleration must be very carefully studied and the mixture-ratios maintained unfailingly accurate. Too little compensation will meet with failure, and too much will produce a waste of fuel.

One other point worthy of note is the frictional loss across the carbureter. Probably the worst offender in cutting down the engine power, and hence the efficiency, is the plain-tube and allied types. However, there are some air-valve devices that show a high frictional-loss. This is illustrated in a comparison of two plain-tube types and an air-valve type, all of 1¼-in. size, shown in Fig. 39. The first plain-tube carbureter has a very high loss before the metering characteristics break-down; the air-valve type is very much lower but, even so, the loss is too high for this type and the capacity is rather low with respect to metering. The other plain-tube carbureter is lower than the air-valve type, even though it is a plain tube, and the metering characteristics are just beginning to show an enrichment toward the break-down point. The frictional loss is critical just above 120 cu. ft. per min., which would limit the capacity.

The causes of the high fuel-waste due directly to the carbureter are: (*a*) improper mixture-ratios; (*b*) poor acceleration, (*c*) omission of temperature control, and (*d*) high fluid frictional-loss.

With cars operating at ordinary speeds, the percentage of engine load is small and requires a high manifold-vacuum; this results in a large engine pumping-loss. The leaner mixtures allow a higher compression for the same power development, which reduces this large pumping-loss. If little or no provision is made for acceleration, the mixture must be made so rich that no real economy at constant-speed operation can be obtained. In a practical device that fulfills the necessary requirements, the acceleration feature is so important that no operation could be effected without it; in fact, the metering device and other features might well be built around it.

The frictional loss across the carbureter is much higher

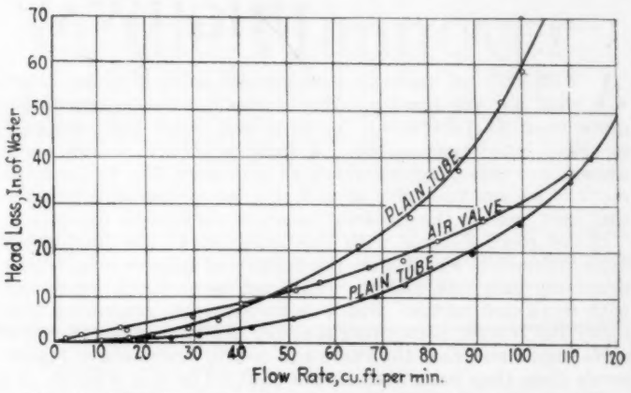

FIG. 39—THE FRICTIONAL LOSS THROUGH CARBURETER BODIES SHOWING THE RELATIVE POWER LOSSES

than through the manifold, and this makes it essential to cut this loss to a minimum. Perhaps a little standardization of the flow-rate capacities and head losses in connection with the carbureter flanges would aid materially in eliminating fuel waste.

The question of temperature and the resulting effect on the metering characteristics is one that needs more research for its absolute determination. Each individual device has its own peculiarities, and some simple automatic control should compensate for summer and winter operation; if preheated air is used, there is all the more necessity for a temperature compensation. Also, engine temperatures on starting and for warming must be included. It is practically impossible to expect 10,500,000 drivers to make their own dash-adjustments for this temperature correction, and the fuel loss is too great to allow inattention.

CONCLUSION

On retrospection, we note the short period of time for the marvelous development of the throttling internal-combustion engine, and, on account of its flexibility, we cannot do otherwise than consider it the automotive engine of the future. In so doing we must realize that now is the time for increasing the expansion-ratios to allow high engine-efficiency, and good carburetion must be effected whereby that high engine-efficiency can be realized to give the maximum return in ton-mileage per gallon. Even though the fuel does not decrease in volatility to the extent previously mentioned, from the standpoint of the motoring public and from an economic viewpoint, carburetion, as applied to the present-day equipment, is far behind the other component parts of the engine. More effort is being expended each year on carburetion development, and the day is at hand when the United States must efficiently utilize more than 75 per cent of its annual consumption of gasoline.

We wish to acknowledge our appreciation to J. W. Geiger, H. A. Huebotter and Opie Chenoweth of the Purdue University Engineering Experiment Station for their assistance in conducting the tests and computing the results of the various investigations presented in this paper.

HIGHWAY ECONOMICS

A SURVEY of highway development as it is today shows that we are appropriating annually sums aggregating more than $1,000,000,000 for city and rural highway construction and maintenance. A further survey of our needs shows that this program must be continued for 10 years or more if we are to arrive at a connected system of highways that will render the greatest possible service to the public.

If we were dealing with this question in the untroubled days before the world war, the matter of finance would be a relatively easy one and we would not be so much concerned with ways and means. But it is necessary to remember that today the Nation is carrying a heavy tax-budget and, what is more important, that the State and county levies are at higher levels than they have been in the past.

We must weigh well all of the elements entering into any appropriation for any purpose and only those must be made that we know will surely return a direct profit to the public. Further, we must seek to allocate definitely those profits so that the burden will be borne in as large a measure as possible by those who are the beneficiaries.

Broadly speaking, we all agree that those who are alive today and those who come after them down to the children of the third and fourth generations, will benefit from highway construction, provided that, once constructed, the highway is maintained.

The cost of highway construction should be divided accordingly, which, of course, at once implies long-term highway bond issues. Engineers tell us that from 40 to 50 per cent of the cost of construction of a highway is a permanent investment, assuming that the highway has been properly located. The right of way, the grade, the drainage, bridges, shoulders and engineering costs need be paid only once. It is the surface that wears. Engineering records show that here the loss is less than is popularly supposed since in the rebuilding of a surface it is possible to salvage much of the material and use it for a base for new surfacing.

Considering the improvement of the main State systems, which will average about 7 per cent of the total mileage of this Country, this system, when completed, will place all of us on main roads, or but a few miles away; and through the feeder lines we will be enabled to reach the great highway and deliver our commodities to market.

We are proposing that this generation shall undertake to build these roads, charge the cost to capital account, pay for them by long-term bonds, use them and pass them on to the next generation in as good or better condition than we built them. We propose that each generation shall pay a fair rental charge for the use of these roads in the sum of amortization charges and interest, plus maintenance. Interest on bonds and amortization should be paid for from general taxation. Maintenance costs should be charged against traffic. Maintenance should include all charges up to and including reconstruction, save extensions or added replacement costs where an inferior is replaced by a superior type of surface. Only that type of road should be constructed which is adequate for the needs of present and future traffic, as determined by careful surveys of both traffic and economic possibilities.—A. J. Brosseau.

FARM AND URBAN INDUSTRIES

IF the farm production of the Country be taken as a whole, our agricultural producers are receiving for their work but a trifle more in money than they received before the war and in purchasing power, which is the essential thing, less than 70 per cent of their prewar remuneration. In other words, their productive operations are completely unprofitable and they have every incentive to better their economic condition by abandoning the farms in favor of those industrial and urban occupations that are visibly several times over superior to their own. And insofar as the farm wage-earners, as contrasted with the farm-owners, are concerned, this incentive is rendered more powerful by the fact that the farm-owner cannot afford, with the prices of agricultural products what they now are in both an absolute and a relative sense, to put wage-costs into what they produce; so that in many sections of the Country we have the curious spectacle of a greatly reduced supply of farm labor that is nevertheless redundant even at very low wages. Thus reports from all parts of the Country recently obtained by the Department of Agriculture showed an excess of the supply of farm labor over the demand in respect of 36 per cent of the reporting districts, an even balance of supply and demand for 31 per cent of the reporting districts, and a slight excess of the demand over the supply for only 33 per cent.

The Country is entering upon a vast process of economic readjustment in which its agricultural production will decrease relatively to a sufficient extent to enable the forces of supply and demand to bring the prices of farm products in general into a proper balance with the prices of industrial products. Speaking from the standpoint of economic law, it will be through this process of readjustment, necessarily painful to all concerned, and not through increased credit facilities and the like, that the disadvantages from which the farmers now suffer, and about which they inevitably complain, will in the end be rectified.—A. R. Marsh in *Economic World*.

ABRASIVE QUALITIES OF STEELS

MODERN equipment is designed for high speed and heavy duty. The manufacturers have met this condition by using heat-treated plain carbon and alloy-steels in the moving parts, such as gears and cams. Application of various types and grades of steel has brought forward the fact that any one steel cannot be used indiscriminately but must be studied in relation to the service for which it is intended. An introductory study of the effect of heat-treatment on the abrasive qualities of plain carbon and alloy-steels indicates that the greater the density and hardness are, the greater the abrasive resistance will be; the ordinary tension and Brinell tests are not a reliable index to the abrasive qualities of a material; the carbon-steels tested showed more uniform results than the nickel-chromium steels and less average wear-loss; and an extensive series of additional tests will have to be made before it will be possible to determine with any degree of certainty the combined effect of ductility and hardness on the abrasive properties of materials.—From a paper by A. M. Cox presented at a recent meeting of the Pittsburgh Section of the American Society for Steel Treating.

Vol. XII January, 1923 No. 1

The Development of the Hannover Sailplane

By Georg H. Madelung[1]

Cleveland Section Paper *Illustrated with* Diagrams

KEY TO FORMULAS AND SYMBOLS

The symbols and formulas given below relate to all of the charts reproduced as well as to the text of the paper, and are grouped in this manner for ready reference by the reader:

A = Area of Wing.
$A_1 = \Sigma(A_p K_x)$ = Apparent Parasitic Area
A_c = Circumscribed Area of Torque Tube
A_p = Parasitic Area
a = Distance of Beam Axis behind Leading Edge
b = Height of Beam Axis above Chord
c = Mean Chord
$c_1 = \Sigma(K_y c) = L/sq$ = Apparent Chord
$D = D_p + D_i + D_s$ = Total Drag
D_i = Induced Drag
D_p = Parasitic Drag
D_s = Section Drag
FS = Factor of Safety
$f = D_s/L$ = Section Drag Divided by Lift ~~Ratio~~
γ = Gliding Angle; $\tan \gamma = D/L$; $\sin \gamma = w/v$
k, k_1 = Factors Used in the Theory of Induced Drag
$K_m = M/qAc$ = Coefficient of the Moment about the Leading Edge
K_{m_1} = Coefficient of the Moment about the Beam Axis
$K_n = K_y \cos \gamma + K_x \sin \gamma$ = Coefficient of Normal Force
$K_T = K_x \cos \gamma - K_y \sin \gamma$ = Coefficient of Tangential Force
K_x = Drag Coefficient
$K_{xs} = D_s/Aq$ = Section-Drag Coefficient
K_y = Lift Divided by the Area of Dynamic Pressure = Lift Coefficient
L = Lift
M_t = Torque Moment about the Beam Axis
$q = \rho(v^2/2)$ = Dynamic Pressure
ρ = Density of Air
s = Span
t = Shear-Strength
th = Wall Thickness
v = Forward Speed
W = Weight
w = Sinking Speed = Rate of Descent

AFTER distinguishing between airplane-racer and sailplane problems, the author discusses the soaring problem briefly and concludes that the outstanding requirement is a minimum rate of descent, since this is the factor that determines the merits of the sailplane. He mentions some features of the early gliders, and then gives an analysis of the rate of descent, subdividing the drag into the three factors of parasitic, induced and section drag, and analyzing each separately with the aid of diagrams.

The span, the wing section and the best chord receive similar treatment. Reasons are given why it was not deemed advisable to build a biplane, the total amount of the rate of descent is determined and the total gliding-angle is subdivided into its proper parts of parasitic, induced and section drag. After stating why external bracing was not used, the form and arrangement of the sailplane body are described and tests of a model are commented upon, together with a brief discussion of the results of some static tests on a piece of the beam. A detailed analysis of torque and the torque moment is included. The radical departures from the conventional practice that were incorporated in this sailplane are stated to be the football wheels, a body of unusual form and arrangement, the large span, the single beam and the torque-carrying plywood-cover.

IT has been stated, in regard to the difference between racer and sailplane problems, that the racers which competed for the Pultizer trophy at Detroit are: "The opposite extreme, in tremendous power applied to the airplane, from the gliders with no power that have been flown recently in Germany." That statement is very true, and it remains true when it is applied to the construction of the wing or any other part. In the case of the racer, large forces are concentrated upon a compact structure; in the case of the glider, small forces only are concentrated, but they are carried by a spacious structure. The victorious Curtiss Army racer had 1950 lb. of gross weight and a 19-ft. span; the Hannover sailplane has only 430 lb. of gross weight, but it has a 41.3-ft. span. Therefore, the structural problems are different.

The same extreme difference appears also in the aerodynamic problems. The victorious Curtiss racer performed at a racing speed that was more than three times its landing speed. It had, therefore, to show its merits at an extremely low lift-coefficient, which is only about 10 per cent of the coefficient of its maximum lift. Two opposite cases had to be considered:

(1) Low angle of incidence; very little lift required; and all effort concentrated to the reduction of drag
(2) High angle of incidence; lift as large as possible; and drag of no importance

It is clear that two requirements that are so different must handicap each other. No such handicap exists in the case of the glider. Its maximum performance is only 10 to 20 per cent more than the landing speed. The case is that of a high angle of incidence, a high lift and a drag that is as low as possible. The characteristic difference between the problems of the racer and those

[1] M.S.A.E.—Project engineer, Glenn L. Martin Co., Cleveland.

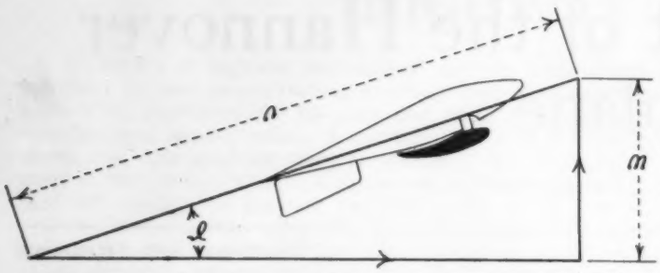

FIG. 1—DIAGRAM ILLUSTRATING THE RELATION EXISTING BETWEEN THE SINKING SPEED OF A SAILPLANE AND THE RATE OF TRAVEL THROUGH THE AIR

of the glider must be recognized clearly. The knowledge of the problem shows the way to its solution.

In the last Rhoen meetings, the highest rewards were offered to those sailplanes that would remain the longest time aloft and rise above the starting point. It seemed to be easy to rise above the starting point if only the up-current were strong enough. However, this result had never been realized. Flights of several minutes' duration had been made but with a considerable loss of altitude. None of the gliders had been able to continue to soar at the same altitude for even 1 min., and Orville Wright's record remained unbroken.

In April, 1921, when I was asked by Professor Proell to design a sailplane for the Hannover Institute of Technology, I was possessed with two ideas. One was that of a biplane with a very thin wing of low minimum drag. The other was that of a monoplane of high wing-loading, having slotted wings. The slots were supposed to be open for the landing condition and to bring a high lift, with a very bad drag, of course; for higher speed, the slots were to be closed. I discarded both ideas, although they would have been useful if a large ratio of speeds had been required, as in the case of the racer. In our case, however, the outstanding requirement is *a minimum rate of descent.*

The rate of descent is the factor that determines the merits of the sailplane. To soar means to sink more slowly than the wind rises. A good pilot is able to acquire power from windgusts. This may be 1 per cent of the total required power, 10 per cent or perhaps more. We know so little about the turbulence of wind that I do not know what the percentage of power acquired in this way would be. But we know the sinking speed of the airplane, and we know how to influence it.

In the early gliders, not much had been done to reduce the rate of descent. The effort of the designer was to make the glider as light as possible, and to reduce the landing speed. The extreme of this tendency was exemplified in the Pelzner glider, a biplane having an area of about 180 sq. ft. and carrying 50 lb. of dead-load. Such a very small amount of dead-load is desirable in the case of a glider that is without landing-gear, where the pilot starts with and lands on his own feet. To fly such a glider is a wonderful physical performance; maybe it is even a better sport than soaring in a sailplane. But the flying performance of the Pelzner glider was exceedingly poor. In 1920, the Klemperer glider was developed by the Aachen Institute of Technology. It was built like a real airplane and followed the sound principles of conventional aircraft engineering. Piloted by Klemperer, it won all the first prizes at the 1920 meeting. However, the rate of descent had not been given first consideration in its design; it made fine flights, but it did not succeed in soaring.

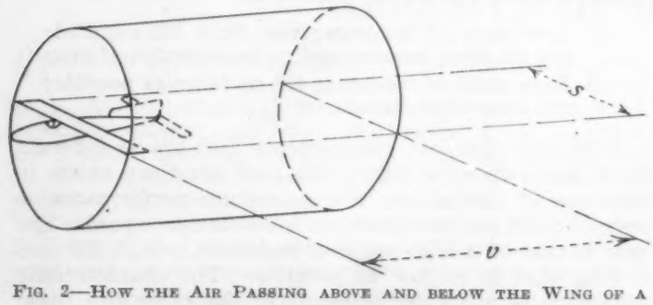

FIG. 2—HOW THE AIR PASSING ABOVE AND BELOW THE WING OF A SAILPLANE PRODUCES LIFT

ANALYSIS OF THE RATE OF DESCENT

Let us determine by what means a low rate of descent can be obtained. With reference to Fig. 1, the sinking speed is the vertical component of the travel through the air and $w = v \sin \gamma$. So long as the gliding angle γ is small, it can be approximated by $w = v \tan \gamma = v\,(D/L)$. It appears that a good gliding-angle or a low drag, and also a low speed, contribute to our purpose. In considering how these values are determined, it is necessary to divide the drag into (*a*) parsitic drag, (*b*) induced drag and (*c*) section drag.

The conception of the parasitic drag is sufficiently well known. It is produced by the non-lifting parts such as the body, the landing-gear, the tail-skid, the tail surfaces and the like. We have

$$D_p = \Sigma(A_p K_x)\, q$$

In abbreviated form,

$$D_p = A_1 q$$

The induced drag, that part of the drag of a wing which is induced by the lift, was introduced by Professor Prandtl of Goettingen and his co-workers. The idea can be explained by the following approximation. In Fig. 2, a certain mass of air passes above and below the wing, and it is used to produce lift. Therefore, $Q = \rho\, K s^2 v$. To produce the lift, it must receive the downward movement behind the wing, and $W_1 = L/Q = L/\rho\, K s^2 v$. Under the wing, the current has already reached part of this downward movement. An analogy is that the air yields under the wing like loose sand under one's feet. The larger the shoes one wears and the faster one walks, the less the sand will yield. It is tiring to walk through loose sand. With each step, one must climb from the hole the other foot has made. This same effect applies also to the wing. It produces its own down-current and, to maintain constant altitude, it must rise constantly in its own down-current. This rising produces the induced drag and

$$D_i = L\,(w/v) = L^2/k_1 \rho s^2 v^2$$

The value of k_1 can be determined by experiment. Prandtl determined it by the methods of classic hydrodynamics and checked the result by experiments. He found that $k_1 = 0.5\,\pi$ for the monoplane; $k_1 = 0.6\,\pi$ for the biplane, with a gap-span ratio of about 1 to 10; and $k_1 = 0.65\,\pi$ for the biplane, with a gap-span ratio of about 1 to 6. This means that a biplane can utilize a little more air to produce lift than a monoplane having the same span, and, the greater the gap is, the more air it can use. Therefore, for the monoplane, $D_i = L^2/\pi\, s^2 q$, and, for the biplane with a gap-span ratio of 1 to 10, $D_i = L^2/\,1.2\,\pi\, s^2 q$.

The conception of section drag is assisted by reference to Fig. 3. The induced drag would be the drag of the ideal wing, with ideal distribution of lift and no skin friction. We find, however, from experiments, that an

additional drag exists, due to the wing imperfections. An idea of the origin of this trouble can be obtained by looking from the deck of a steamer down to its waterline. The skin of a ship is covered by a sheet of heavily turbulent water, 2 or 3 in. thick on the sides and dissolving at the rear into a deadwater, a broad band of vortexes. The same effect occurs on wings, and the drag that it produces is called section drag; so,

$$D_s = K_{xs} A q$$

or, in another form,

$$D_s = fL$$

The second form of the section-drag equation is the more useful for our purpose. The factor f can be considered as a friction coefficient, a measure of the imperfectness of a section. The factor f is, of course, not a constant; it is different for every angle of attack. In a racer, a wide range of angles must show a low section-drag; in our case, we have no such handicap and consider only the minimum value of f. In many sections, we find $f_{min} = 1/40$; frequently, $f_{min} = 1/50$; occasionally, $f_{min} = 1/80$.

Having completed the subdivision of the drag, we can now write the formula for the total drag. It is

$$D = D_p + D_i + D_s$$

or,

$$D = (L^2/\pi s^2 q) + A_1 q + fL$$

But, in this form, the formula is not yet useful to show the influence of the dimensions of the sailplane upon the performance. To show this influence, the dynamic pressure q must be eliminated and be replaced by the lift, L, the apparent chord, c_1, and the span, s.

Since $q = L/sc_1$, we obtain the formula for the gliding angle. For the monoplane, it is

$$\tan\gamma = D/L = A_1/c_1 s + c_1/\pi s + f$$

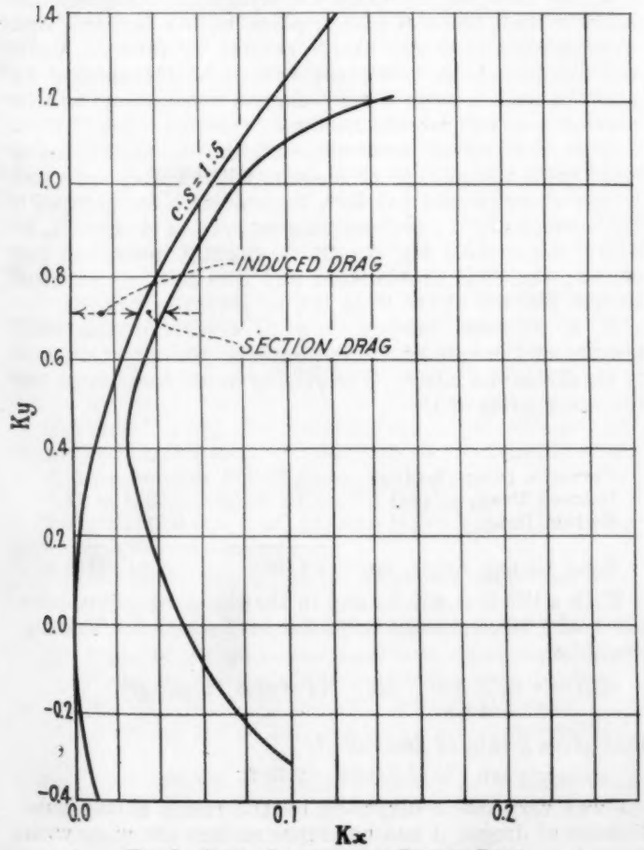

FIG. 3—CHART ILLUSTRATING SECTION DRAG

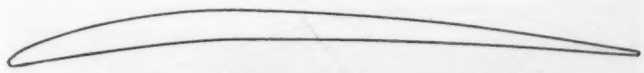

FIG. 4—THE GOETTINGEN WING SECTION NO. 427

For the biplane having a gap-span ratio of 1 to 10, it is

$$\tan\gamma = A_1/c_1 s + c_1/1.2\pi s + f$$

The next step is to express the speed, v, when using the same values. We have, $v = \sqrt{(2L/\rho c_1 s)}$ and, by introducing it into the equation, we obtain the rate of descent

$$w = wv \tan\gamma$$
$$= [\sqrt{(2L/\rho)} \times \sqrt{(1/c_1 s)}]\,[(A_1/c_1 s) + (c_1/\pi s) + f]$$

for the monoplane, and we can write a corresponding formula for the biplane. This formula is very useful. It contains no factors other than the well-known density of the air and a few dimensions of the craft such as L, the weight or lift; c_1, the apparent chord; s, the span; A_1, the apparent parasitic area; and f, the "friction" coefficient of the section. These craft dimensions can be estimated or determined easily.

It is remarkable how large a function the span has upon the rate of descent as expressed in the foregoing formula; s is found below the line only. An increase of span must, therefore, reduce the sinking speed. It was for that reason that I insisted on giving the sailplane the largest span that we believed we could handle. The importance of the amount of span has not been fully understood outside, even by experts. For instance, the editor of *Flight* speaks about "the wing span, which is obviously somewhat large," on Aug. 31, 1922, after the record was broken. I even had difficulty to obtain permission to use such a large span as I did use. To quote from my letter of April 30, 1921, I had to explain that

> If we reduce the span, the induced drag will increase. It is $D_i = L^2/\pi s^2 q$. In our case we estimate that the lift, L, is 310 lb.; the span, s, is 41.3 ft.; and the dynamic pressure, q, is 1.28 lb. per sq. ft. If we have to reduce the span to the railroad limit, 29.5 ft., the drag will increase by 13.4 lb., as follows:
>
> $$\Delta D_i = (310)^2/1.28\pi[(1/[29.5]^2 - 1/[41.3]^2)] = 13.4 \text{ lb.}$$
>
> That is considerable; it increases our rate of descent by
>
> $$\Delta w = v(D_i/L) = 1.42 \text{ ft. per sec.}$$
>
> and it is just this value which determines whether we can compete.

The large span was granted at last. Its success was so convincing that, in the following year, out of 53 entries, 33 had a span of 36 ft. or more; 19, a span of 40 ft. or more; and 11, a span of 45 ft. or more. One, a monoplane built by Espenlaub, a young village carpenter, had even a 55.8-ft. span and, in spite of a bad finish, showed a very good performance.

SECTION AND BEST-CHORD DETERMINATION

The next thing to determine was a good section. I wished to find a section with a low D_s/L ratio for f. The Goettingen section No. 427, illustrated in Fig. 4, is very good. At $K_y = 0.95$, it has a ratio of 1/87 only. However, it is very thin. For structural and for service reasons, I wished to use a thick section. Also, the section in Fig. 4 had rather low lift-values, and would have required more wing area than I wished to use. Of the thick sections available, Goettingen section No. 441 looked promising. It showed a very high lift and a good ratio, about 1/50 over a wide range and especially for the high lift-values, with a minimum near $K_y = 1.0$. It was almost thick enough for internal bracing, but it was desirable

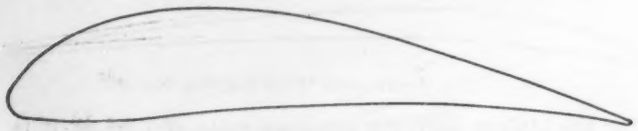

FIG. 5—THE COMPOSITE WING SECTION USED IN THE HANNOVER SAILPLANE

to make it a little thicker anyhow. To save time, Dr. Betz, of Goettingen, advised us to form a new section by combining the existing templates of Goettingen sections No. 384, upper side, and No. 441, lower side. The section No. 482 so obtained was tested and developed to be very satisfactory. It had almost the same properties as Goettingen section No. 441. The maximum thickness is 18 per cent of the chord. Such an ample thickness permits the construction of a very stiff, internally braced wing of sufficiently low weight. It is shown in Fig. 5.

The next problem to solve was that of finding the best chord. This means the chord that will give the minimum rate of descent, w. The best value of c_1 is found by differentiating w for c_1 and setting $dw/dc = 0$. This gives, for the monoplane,

$$c_{1\,best} = 0.5\pi[(fs \pm \sqrt{\{f^2s^2 + (12A_1/\pi)\}}]$$

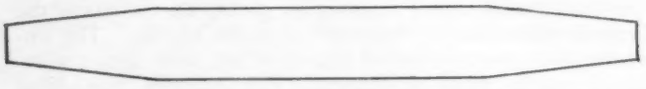

FIG. 6—PLAN VIEW OF THE WING USED IN THE HANNOVER SAILPLANE

For the biplane having a gap-span ratio of 1 to 10,

$$c_{1\,best} = 0.6\pi[(fs \pm \sqrt{\{f^2s^2 + (10A_1/\pi)\}}]$$

The chord that gives the best gliding-angle is smaller. For the monoplane

$$c_{1\,best} = \sqrt{\pi A_1}$$

For the biplane with a gap-span ratio of 1 to 10

$$c_{1\,best} = \sqrt{1.2\pi A_1}$$

These formulas for the best chord are surprisingly simple. They *contain* nothing but three values that can be estimated or determined in the early design; the span, s; the D_s/L ratio, f; and the apparent parasitic area, A. We had the following values:

s, ft.	41.3
f	1/55
sf	0.75
Cross Area of the Body, sq. ft.	6.5
Estimated Drag Coefficient	0.20
Area of Tail Surfaces, sq. ft.	34
Estimated Drag Coefficient	0.012
$A_1 = (6.5 \times 0.20) + (34 \times 0.012)$	1.71

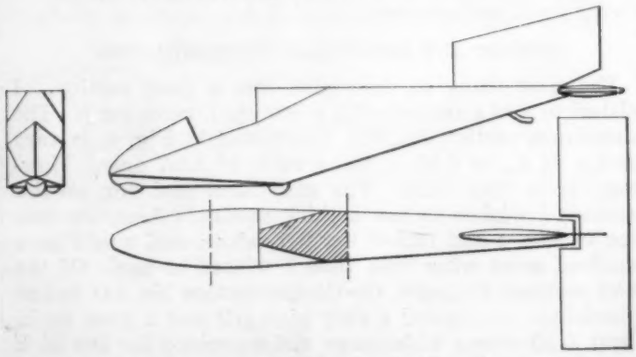

FIG. 7—END AND SIDE ELEVATIONS AND PLAN VIEW OF THE BODY OF THE HANNOVER SAILPLANE

The best apparent chord of the monoplane is, therefore.

$$c_{1\,best} = 0.5\pi[0.75 \pm \sqrt{\{(0.75)^2 + (12/\pi \times 1.71)\}}]$$
$$= 5.35 \text{ ft.}$$

For the best gliding-angle,

$$c_{1\,best} = \sqrt{1.71\pi} = 2.32 \text{ ft.}$$

A close investigation, details of which are given in the Appendix to this paper, shows that the most advisable chord lies between the so-determined optima. At $c_1 = 4.2$ ft., the rate of descent is only 1.4 per cent more than the minimum, but the gliding angle is 12 per cent better than for $c_1 = 5.35$ ft. At $c_1 = 2.32$ ft., on the other hand, the gliding angle would have been still better but, also, the rate of descent would have been 20 per cent worse.

MONOPLANE OR BIPLANE

A further investigation, as explained in the Appendix, shows that in the case of a biplane a slightly smaller span would be sufficient to bring the same results. With the thick section No. 482, a span of 1 ft. would be spared and, with the very good section No. 427, a span of even 5 ft.

I thought that such a small gain in span would not make the biplane preferable. The biplane requires twice as many beams and many more ribs and fittings, and wires and struts which can be avoided in the monoplane. This consideration applies especially to the very thin section No. 427 that is so very good aerodynamically. I doubt if a wing with this section could be constructed with only one pair of struts on either side.

The investigation showed also that the chord of the biplane would have to be very small, about 2.2 to 2.7 ft. It is certainly not impossible to build such narrow wings; in the case of very large craft, it has even the advantage that the elements of which the wings are composed are handier than those of a monoplane with a larger chord. For instance, they still can be shipped by railroad, while the corresponding monoplane has to be transported by air. In such a case, a very careful comparison will be necessary to balance the merits of the two types.

This analysis is, however, sufficient to show that a more conventional type of biplane, with shorter span and longer chord, would not have been able to compete with the monoplane. I gave the wing an area of $A = sc_{mean} = 41.3 \times 4.2 = 174.0$ sq. ft. It is slightly deeper in the center; the ends are tapered and decrease to the good section No. 427 at the wing tip, as shown in Fig. 6.

It is of some interest now to determine the total amount of the rate of descent and the gliding angle, and to subdivide the latter. The gliding angle consists of the following parts of the

			Per Cent
Parasitic Drag, $A_1/(c_1 s)$	$= 1.71/174$	$= 0.0098$	$= 16.3$
Induced Drag, $c_1/(\pi s)$	$= 4.2/41.3\pi$	$= 0.0324$	$= 53.7$
Section Drag, f	$= 1/55$	$= 0.0182$	$= 30.0$
Total Gliding Angle, $\tan \gamma = 1/16.6$		$= 0.0604$	$= 100.0$

With a lift $L = 430$ lb. and in the standard atmosphere $\rho = 1/422$ (lb. $\times$ sec.^{2}/ ft.^{4}), the best speed for soaring would be

$$v_{best} = \sqrt{(2 \times 430 \times 422)/174} = 45.6 \text{ ft. per sec.}$$
$$= 31.1 \text{ m.p.h.}$$

This gives a rate of descent of

$$w_{best} = v \tan \gamma = 45.6/16.6 = 2.75 \text{ ft. per sec.}$$

I was very much surprised by the result of the subdivision of drags. I had not expected that the wing with the "obviously somewhat large span" would produce so

much induced-drag, 54 per cent of the total drag, and that the parasitic drag would be so little, only 16 per cent. Of course, there were no struts, wires, axles and wheels, but the tail surfaces and the body were large for a small sailplane of only 174 sq. ft. Also, considering the rough form of the body, a large drag coefficient, 0.20, was used in the analysis; good forms have less than 0.06.

In regard to the reason the wing was not braced externally, a streamline tie-rod on either side would not have affected the performance to any extent. I omitted them, not so much for the saving of drag as for manufacturing and service reasons. In my letter of April 30, 1921, I wrote, "If there is the slightest chance, I will try to avoid external bracing, not so much to save drag as for the simpler assembling." It is obvious that wires and wire connections are in constant danger because of the rough handling that a glider must withstand during transportation over a rolling country.

Form and Arrangement of Body

The body is shaped somewhat oddly, as indicated in Fig. 7 and shown in Fig. 8. No effort was made to give it streamline forms. It is square in the rear, ending in a horizontal knife-edge. It is hexagonal in the front, with straight sides and sharp edges running from front to rear; and they even do not run steadily, but are broken below the wing. The lower break is below the center of gravity to allow a 16-deg. tilting fore-and-aft, as wheels do. The upper break occurs at the trailing edge of the wing and, from there, the upper longerons run straight down to the nose to give sufficient free space for the pilot. I determined the form of the body principally for constructional reasons.

One very important point had been considered with reference to the arrangement of the body. A series of experiments in the wind-tunnel of Dr. Junkers, at Dessau, in 1920, had demonstrated that, for the condition of high lift, a body below the wing is much superior to a body on top of it. This is not so important for fast airplanes, flying at double landing-speed or more, but it is of high importance for the very low-power airplanes. It developed that in many cases the drag of the entire airplane is much more than the sum of the detail drags. I took the precaution, therefore, to place the body below the wing; and to leave the entire upper side and the front third of the lower side free from obstructions.

Another lesson learned from Dr. Junkers' experiments was that the near surroundings of the wing must be free from pockets in which the air speed has to decrease. This was carefully avoided. The joint between the body and the wing begins just behind the pilot's head, one-third of the chord from the leading edge, and widens constantly to the full width of the body at the trailing edge.

Model Test

A model test was run at Goettingen. The following arrangements were tested:

(1) Standard wing model, to determine properties of section

(2) Model of the intended wing, tapered, to 1/10 scale

(3) Model of the entire sailplane, with the elevator adjusted for every angle of incidence so as to balance the craft about the estimated center of gravity

(4) A similar test to (3), but with a round streamline body

Test (2) confirmed Prandtl's theory of induced drag well, and the results were even slightly better than those transformed from Test (1), as shown in Fig. 9. Test (3) gave the somewhat surprising result that the parasitic

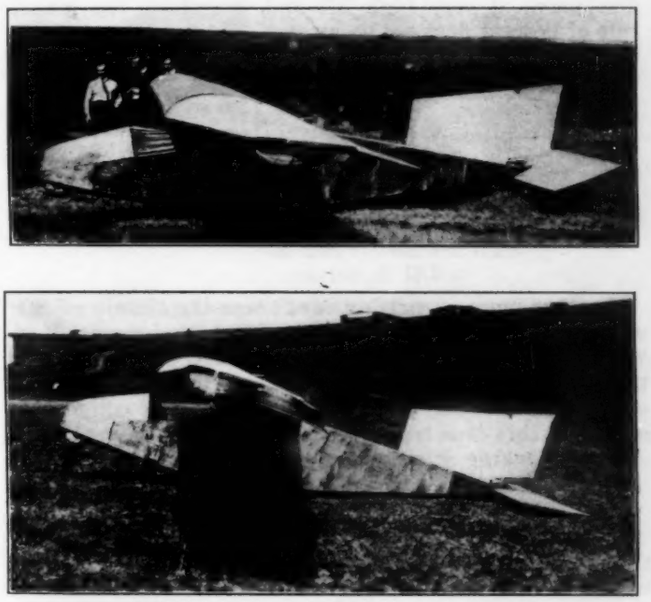

Fig. 8—Two Views of the Hannover Sailplane Showing the Somewhat Peculiar Shape of the Body

drag is not constant. At low angles of incidence, it is very bad. The wing is here very bad also, particularly below $K_y = 0.4$, where a lower burble-point is reached. (See Fig. 10.) On the other hand, the parasitic drag

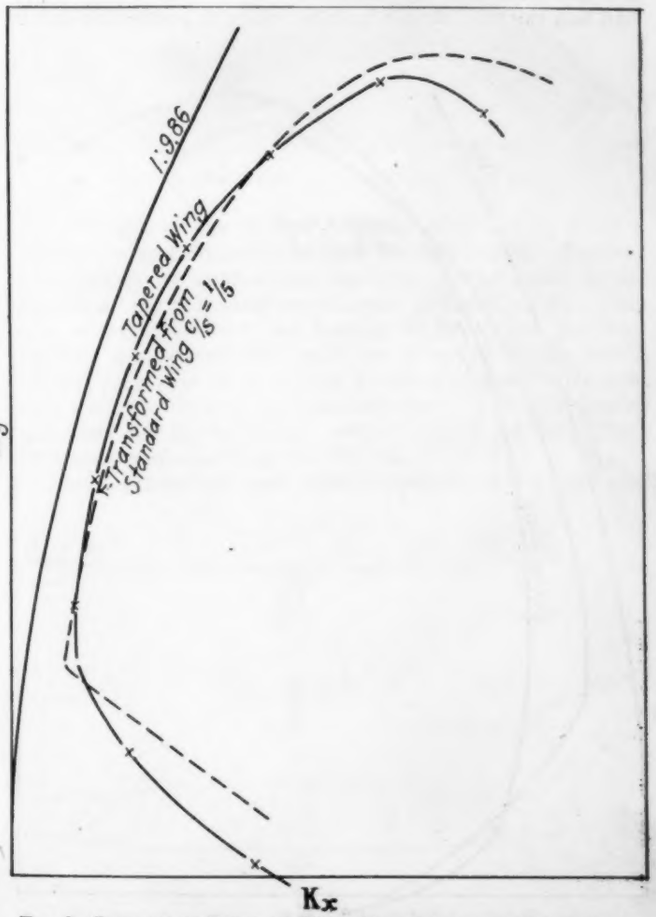

Fig. 9—Results of Tests on Wing Models To Determine the Properties of the Sections

disappears at very high lift and, therefore, the lowest rate of descent does not occur at $K_y = 1.0$, as the analysis had determined, but at $K_y = 1.36$, with a gliding angle $\tan \gamma = 1/15$. This determines the best speed for soaring to be

$$v_{best} = \sqrt{(2 \times 430 \times 422)/(174 \times 1.36)}$$
$$= 39.2 \text{ ft. per sec.}$$
$$= 26.7 \text{ m.p.h.}$$

It determines also that the minimum rate of descent is

$$w_{best} = v \tan \gamma = 39.2/15$$
$$= 2.61 \text{ ft. per sec.}$$

Another more surprising result was the failure of the streamlined body, Test (4), to give better results. It was even slightly inferior for high lift. The only explanation that we could imagine was the more complicated joint between the round body and the wing. At any rate, this is a typical example of costs having been saved by making a laboratory test.

Constructional Details

The question may arise as to why, with the knowledge that the round streamline body of Test (4) did not give better results, we did not discard the body entirely. I am glad we have not discarded it, because it has helped considerably to beat the record. In the Hannover sailplane the pilot is enclosed entirely; nothing but his head protrudes from the body and he is well protected from the wind and cold, as shown in Fig. 11. Hentzen reports that he felt comfortable during his great flight, while the spectators on the ground built fires to warm themselves. We have not had any bad crash, but I believe that if we had had the pilot would have found good protection in the

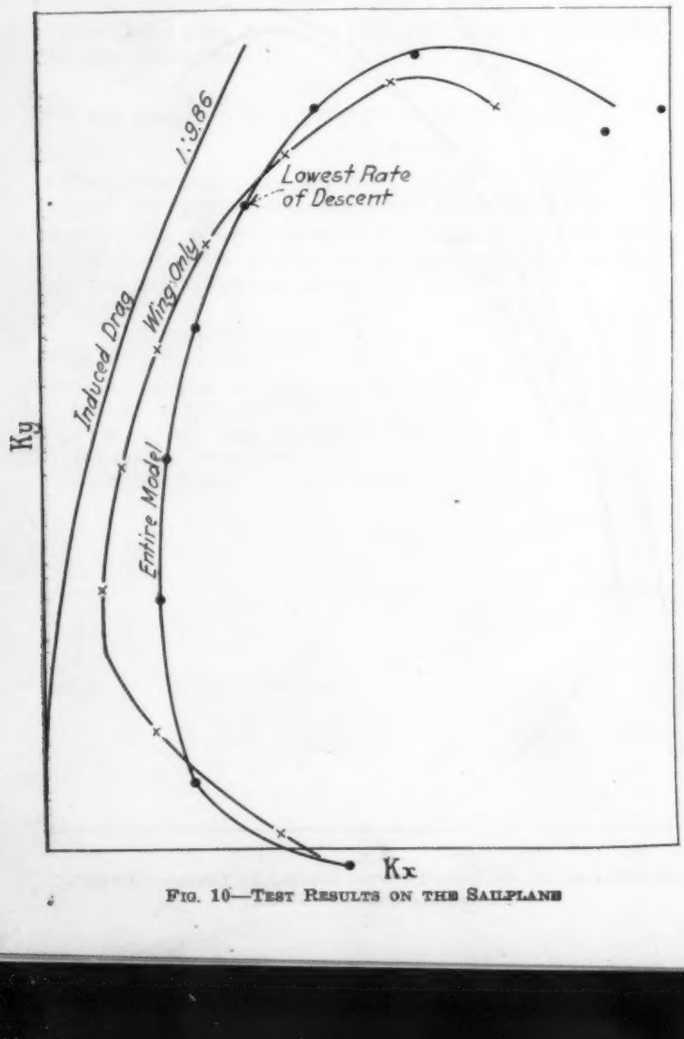

Fig. 10—Test Results on the Sailplane

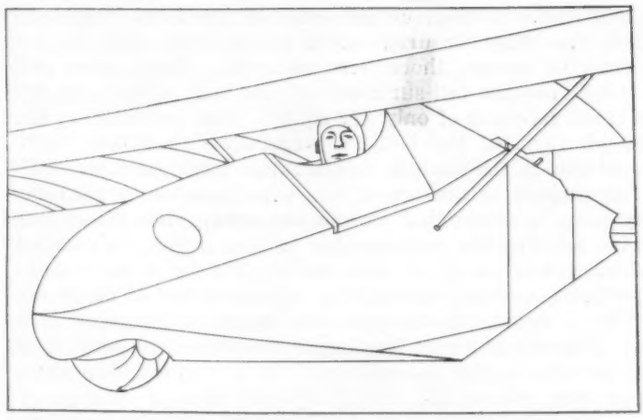

Fig. 11—The Pilot in the Hannover Sailplane Was Well Protected against the Wind and the Cold

body. Plywood bodies are the most crashproof, especially with ash longerons that are wrapped with fabric. Our longerons are laminated and form an uninterrupter bow in the nose; they absorb much energy before destruction and do not splinter so badly as spruce.

A feature which helped very much to avoid crashes was the "football wheel," suggested by Herrmann Dorner, well-known designer of aircraft and chief engineer of the Hannover Wagon Co. This is a rimless wheel in which the inner tube is ball-shaped and rides on the hub; the casing is connected to the flange of the hub. This wheel worked very satisfactorily.

The construction of the wing is somewhat similar to that of Fokker's wing, or its predecessor of 1916, an all-wood cantilever wing designed by Dr. Oesterlen of the Hannover Institute of Technology. It contains no internal wire-bracing or metal fittings; metal is used only where the outer and the center wing join. A three-ply cover of the front third forms the skin and the drag bracing; the rear two-thirds of the wing is made in the usual way, and the ribs are covered by doped fabric. The new feature of this wing is that it does not contain the classic arrangement of a front and a rear beam; it has one single beam only, one-third of the chord from the leading edge, where the section has its greatest depth.

Some constructional problems arise from this new arrangement. The ribs have to cantilever toward the trailing edge for two-thirds of the chord. However, this is for 3 ft. only, which is not more than the rear cantilever part of an ordinary medium-sized rib. Also, as the beam is 11 in. high, the free length is not out of proportion.

Torque

A more difficult problem was how to master the torque that tends to tilt the wing forward at high speed. At low angles of incidence, in all cambered wings, the nose part gets a down-pressure while the rear has still an up-pressure. The single beam is not able to carry the moment that results therefrom. It carries only the forces in its plane. The torque moment turning about the beam is taken by the plywood-tube that forms the cover of the front third of the wing. The value of this moment is

$$M_t = qA\ (-aK_n + bK_t + cK_m$$
$$= qAcK_{m_1} \text{ in abbreviated form}$$
according to Fig. 12

The coefficients K_m, K_n and K_t are taken from the test of the bare-wing model. The value of the dynamic pressure q is found from

$$q = W/[A\sqrt{(K_y^2 K_x^2)}]$$

The coefficients K_y and K_x are taken from the test of the entire model. We find, by the elimination of q, that

$$M_t = (W \times c)\,[\{ - (a/c)K_n + (b/c)K_t + K_m \}/ \sqrt{(K_y^2 + K_x^2)}]$$

This is the formula that determines the worst possible case of a torque moment. The moment can reach very high values, especially for fast airplanes. It has made many wings collapse during a dive. We had very fortunate conditions in the Hannover sailplane regarding the speed limit. At low angles of incidence, below the lower burble-point, the drag is abnormally high. The minimum value of $\sqrt{(K^2_y + K^2_x)}$ is 0.092, which is twice as much as in ordinary airplanes. The speed limit for a long steep dive is, therefore, not more than

$$\begin{aligned} v_{max} &= \sqrt{[2W/\{A\rho\sqrt{(K_y^2 + K_x^2)}\}]} \\ &= \sqrt{(2 \times 430 \times 422)/(174 \times 0.092)} \\ &= 150 \text{ ft. per sec.} = 102 \text{ m.p.h.} \end{aligned}$$

This speed limit has, naturally, never been reached. However, the pilots claim that they have reached a speed of 56 m.p.h. in a dive, which should warn them not to be careless in the consideration of the high-speed condition.

Returning to further consideration of the torque moment, the moment coefficient

$$K_{m_1} = [-(a/c)K_n + (b/c)K_t + K_m]$$

which, in our case, is

$$K_{m_1} = (-0.33\,K_n + 0.10\,K_t + K_m)$$

has its maximum near $\gamma = -4.6$ deg. and is $K_{m_1\ max} = 0.130$. If the sailplane dives for a long time so that it reaches its speed limit, and then is tilted up to the angle of incidence $\gamma = -4.6$ deg., the torque moment about the beam axis becomes

$$\begin{aligned} M_t &= 430 \times 4.75 \times (0.130/0.092) \\ &= 2890 \text{ lb-ft.} \end{aligned}$$

This is the largest moment that has to be carried from the wing into the body and balanced by the elevator or tail surface. The largest torque-moment of each wing near the root is one-half of this value.

Concerning a torque-tube analysis, our knowledge of the torque moment did not allow us to determine the required wall-thickness of the plywood tube without a test. No information could be obtained in regard to the shear-strength of plywood under such conditions as obtain in the nose of the wing. Under ordinary conditions, the shear-strength of plywood is about $t = 2500$ lb. per sq. in. The circumscribed cross-area of the tube is $A_c = 1.1$ sq. ft. A factor of safety of 2.5 seemed advisable. The required thickness under these assumptions would have been

$$\begin{aligned} th_{req} &= (FS \times M)/(2 \times A_c \times t) \\ &= (2.5 \times 1435)/(2 \times 1.1 \times 2500 \times 12) \\ &= 0.055 \text{ in. near the root.} \end{aligned}$$

Toward the tip, the thickness would have tapered down.

Static Tests and Factor of Safety

We made a torque test on a piece of beam with its ribs and the nose covered with 1/16-in. birch plywood. The sample first stood-up very well and showed little deformation. Then the plywood began to wrinkle; it lost much of its stiffness and sheared finally along a rib and the beam. Although the test did not show the full required strength, it gave us valuable information about the way in which such a structure is destroyed. The phenomenon of wrinkling under shear is very much similar to that of a slender strut bending under a longitudinal load. Unfortunately, no second Euler has yet arisen to bring the problem to a simple solution.

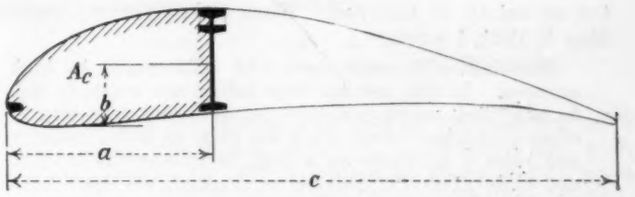

Fig. 12—Diagram Illustrating the Determination of the Torque Moment about the Beam Axis

We brought the ribs closer together to reduce the free wrinkling length, and increased the plywood thickness to 0.08 in. in the center. Toward the wing-tip, it tapers down to 0.063 and to 0.048 in. in thickness. The sample was repaired and another test was made to determine the bending-strength of the beam. It brought no new knowledge; however, it helped us to feel confident about the strength of the wing. A factor of safety of 5 for a uniformly distributed load had been assumed. Another case, which we considered, was that a load equal to the weight of the sailplane would apply to one wing-tip. This case corresponds to a bad landing in which the sailplane touches the ground first with one wing-tip, a case which is not very rare in a rolling country subject to treacherous winds. It was possible, therefore, to lift the loaded sailplane on its wing-tips, with a factor of safety for this abnormal condition of $FS = 2$.

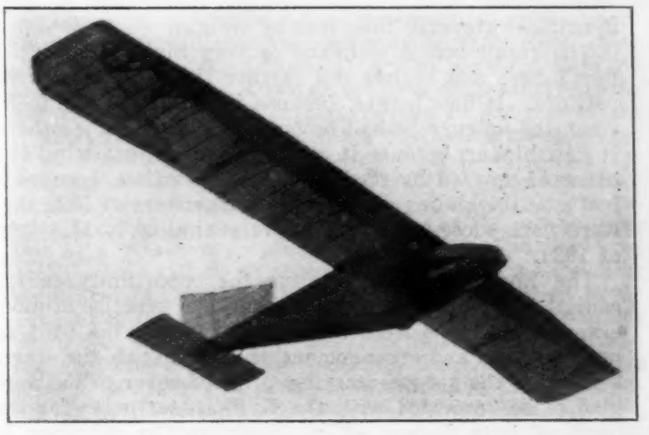

Fig. 13—The Hannover Sailplane in Flight

The question has been raised whether such a high fac-

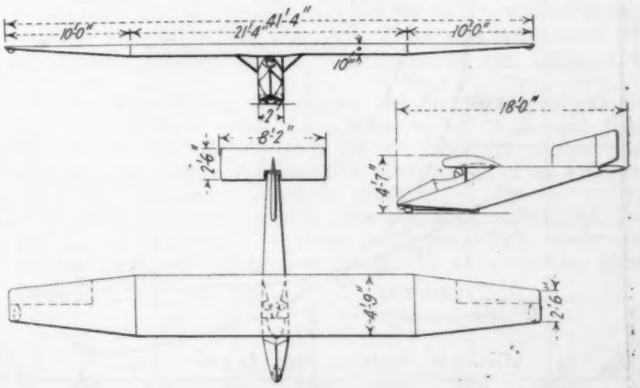

Fig. 14—Plan View with Dimensions of the Hannover Sailplane

tor of safety is justified. When asked for my opinion May 1, 1921, I wrote

> Stunting will come soon and will result in high stresses. It will not be long before air combats will be practised, perhaps not in Germany but certainly in other countries. Also, when we glide in a flat country and raise a sailplane as a kite, to release the towline and glide after the altitude is reached, we shall have to meet large stresses
>
> It is impossible to save much weight by a reduction of the factor of safety; the wing loading that we have to use is too low for that. It is hardly possible to construct any lighter than now. Also, the useful load is such a large percentage of the total load that a saving in the dead-load will not help much

Conclusion

The Hannover sailplane was the heaviest of all the competitors in 1921, at least of those competitors that flew. If all of the features that have been incorporated in it are considered, it will be understood that it could not be very light. But it is evident that the departures from conventional practice brought more gain in aerodynamical property than loss by weight.

The result was a sailplane of very high efficiency; it flew longer and higher and farther than the other competitors. It flew longer, because it could continue to fly when the up-current had become too slow for the others. It flew higher, because it could keep aloft in a wind less deflected upward by the hill. It flew farther, because it had a better gliding-angle. The competitors of 1922 that were next successful were all very similar to this type of 1921.

The Rhoen meeting gave us the opportunity to try radical departures from the conventional practice at moderate cost. These were the football wheels, the body of unusual form and arrangement, the large span, the single beam and the torque-carrying plywood-cover. We were glad to be rewarded with the Kyffhaeuser prize for the wing of best design. Fig. 13 shows the sailplane in flight; and Fig. 14 is a plan view giving dimensions.

Appendix

To find the best chord for the rate of descent, we transform the formula previously given for w and we have

$$w = [\sqrt{(2L/\rho s^2)}]\ [A_1 c_1^{-1.5} + c_1^{0.5}/\pi + fsc_1^{-0.5}]$$

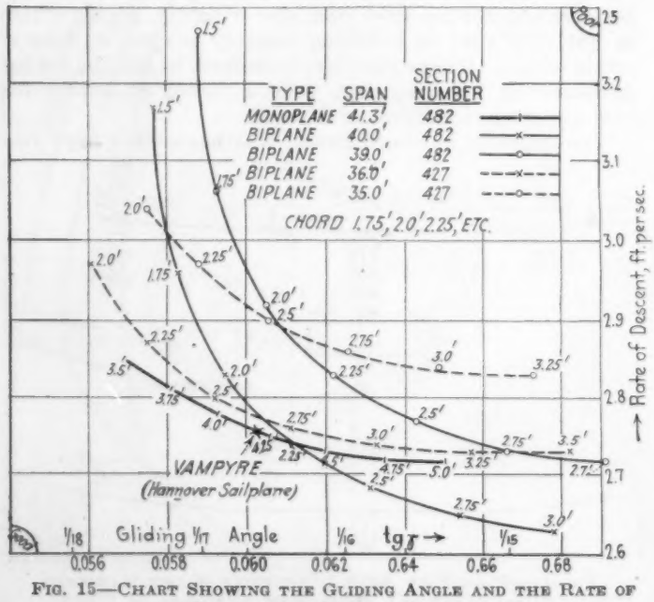

Fig. 15—Chart Showing the Gliding Angle and the Rate of Descent for Various Chords

Differentiating w for c_1 and setting $dw/dc_1 = 0$, we have

$$0 = -1.5\,A_1(c_1^{-2.5}) + 0.5\,\pi\,(c_1^{-0.5}) - 0.5\,fs\,(c_1^{-1.5})$$

We transform the above equation to the form

$$c_1^2/\pi - fsc_1 - 3A_1 = 0$$

Solving this last equation we have, for the monoplane,

$$c_{1\,best} = \pi/2\,[fs \pm \sqrt{(f^2s^2 + 12A_1/\pi)}]$$

For the biplane with a gap-span ratio of 1 to 10, we replace π in the equation just given for $c_{1\,best}$ by 1.2 π. This is the upper limit for the chord.

To find the best chord for the gliding angle, we transform the formula previously given for tan γ and we have

$$\tan\gamma = 1/s\,[A_1(c_1^{-1}) + c_1/\pi + fs]$$

Differentiating tan γ for c_1 and setting $d\ \tan\gamma/dc_1 = 0$, we have

$$0 = -A_1\,(c_1^{-2}) + 1/\pi$$

We transform the above equation to the form

$$c_1^2/\pi - A_1 = 0$$

Solving this last equation we have, for the monoplane,

$$c_{1\,best} = \sqrt{\pi A_1}$$

For the biplane with a gap-span ratio of 1 to 10, we replace π in the equation just given for $c_{1\,best}$ by 1.2 π. This is the lower limit for the chord.

To find the advisable chord for a monoplane, the gliding angle and the rate of descent are determined for a series of chords between 3.75 and 5.00 ft., as follows

The gliding angle

$$\begin{aligned}\tan\gamma &= A_1/c_1 s + c_1/\pi s + f \\ &= 1.71/41.3c_1 + c_1/41.3\pi + 0.0182 \\ &= 0.0414/c_1 + c_1/130 + 0.0182\end{aligned}$$

The rate of descent

$$\begin{aligned}w &= [\sqrt{(2L/\rho\, c_1 s)}]\ \tan\gamma \\ &= [\sqrt{(2 \times 430 \times 422)}/41.3c_1]\ \tan\gamma \\ &= (93.5/\sqrt{c_1})\ \tan\gamma\end{aligned}$$

The values for the series of chords are given in Table 1.

TABLE 1—VALUES OF THE GLIDING ANGLE AND THE RATE OF DESCENT FOR A SERIES OF CHORDS

Chord of Wing		c	3.75	4.0	4.25	4.5	4.75	5.0
Apparent Chord		c_1	3.75	4.0	4.25	4.5	4.75	5.0
Drag/Lift	Parasitic	$0.0414/c_1$	0.0110	103	97	92	87	83
	Induced	$c_1/130$	0.0289	308	327	347	366	385
	Section	0.0182	0.0182	182	182	182	182	182
Gliding Angle		$\tan\gamma$	0.0581	593	606	620	635	650
Rate of Descent		w	2.81	2.78	2.75	2.73	2.72	2.72

The results in Table 1 are plotted in Fig. 15, $w/\tan\gamma$. A chord c = 4.2 ft. gives a good result for both w and tan γ, and is still structurally reasonable.

Monoplane and Biplane Comparison

The results with the monoplane are compared with those of biplanes, as follows. The biplanes have a smaller span and either the same wing-section No. 482, in which $f = 1/55$, or the thin section No. 427, in which $f = 1/87$. They have an additional parasitic-drag due to the struts and wires. We have

ΔA_1 = Number x Length x Thickness x Drag Coefficient

This gives a total parasitic-drag area of

	Sq. Ft.
$A_1 = A_1\,0 =$ (6.5 x 0.20) + (34 x 0.012)	= 1.71
$+\Delta A_1 =$ + 8x 3.5 x 1/20 x 0.27	= + 0.38
= — 8 x 13 x 1/150 x 0.40	= + 0.27
	2.36

Hence, A_1 = 2.36 sq. ft.

The performance values of the biplane are found in the same way as they were determined for the monoplane. For instance, the one with Section No. 427 has the following values

$$f = 1/87 = 0.0115$$
$$K_y = 0.9$$
$$c_1 = 2 \times 0.9 \times c$$
$$s = 36 \text{ ft.}$$

We have, for the gliding angle

$$\tan \gamma = A_1/c_1 s + c_1/1.2\pi s + f$$
$$= 2.36/36c_1 + c_1/1.2\,\pi \times 36 + 0.0115$$
$$= 0.0656/c_1 + c_1/136 + 0.0115$$

For the rate of descent, we have

$$w = [\sqrt{(2L/\rho c_1 s)}] \tan \gamma$$
$$= [\sqrt{(2 \times 430 \times 422)/(c_1 \times 36)}] \tan \gamma$$
$$= (100.3 \times \tan \gamma)/\sqrt{c_1}$$

The results are plotted also in Fig. 15, together with those of the monoplane, $w/\tan \gamma$. It develops that with the thin section No. 427, a biplane of about 36.2-ft. span and having a 2.65-ft. chord would give the same gliding-angle *and* the same rate of descent. With the thick-wing section No. 482, used in the monoplane, a span of about 39.2 ft. and a chord of about 2.15 ft. would be required.

CLOSED-CAR DEMAND INCREASING

THE marked trend toward an increased use of closed cars is indicated by the replies to a questionnaire recently sent out by the National Automobile Chamber of Commerce. This was circulated among 20,000 car-owners and 60 per cent of the replies stated the buyer's next purchase would be an enclosed model. The principal reasons given in the replies were protection from the weather, general comfort, all-year use and a better appearance. In Massachusetts, Illinois, Wisconsin, Iowa, California and Pennsylvania the vote for the closed car was well over 60 per cent while in the Southern States the demand was less marked, only 40 per cent of the owners in Alabama, for example, being in favor of the closed car.

A large number of the replies gave several reasons for the owner's preference. Protection from the weather was listed as one of the factors in 44 per cent of the replies, although general comfort, which ranked next with 34 per cent of the replies, includes not only protection from the weather but also freedom from dust as well as easy riding qualities. In mentioning economy, which represented the main reason in 20 per cent of the replies, some of the owners emphasized the point that in addition to being able to use the enclosed car throughout the entire year, the expense of extra heavy clothing needed for the open models for use in the winter time was eliminated. The inconvenience of side curtains was mentioned by 4 per cent of the owners preferring closed cars as the reason why their next purchase would be a closed vehicle.

On the other hand, the answers to the questionnaire revealed several strong factors that will keep the open models in popular favor. These include poor roads which was mentioned by 23 per cent of the replies as being an objection to closed cars; the price differential, lightness, safety, more air and business utility. Although the price differential is much less now than it was 2 or 3 years ago, 16 per cent of the replies gave this as the reason for purchasing an open type of automobile. Lightness with a resulting lower upkeep cost was emphasized by 15 per cent. Closely following these two factors was the unforeseen argument of safety. In the replies favoring the open car 14 per cent of the owners gave this as a reason expressing a fear of cuts resulting from broken glass and an objection to being closed in in case of accident. The fresh-air advocates were represented by 13 per cent of the replies and 10 per cent emphasized the utility of an open car for business use. The replies from the farming districts in particular indicated that the phaeton was more of a utility vehicle as it can be readily used for cartage and it is also possible to drive it over fields with less inconvenience than would be the case if an attempt were made with a closed car.

FRENCH TEST OF PRODUCER-GAS TRUCKS

FIRST prize in the French gas-producer trials has been won by the British entrant, the Thornycroft suction gas vehicle. The six competing trucks used four-cylinder engines designed to run on gasoline. Four of the trucks employed charcoal as fuel; the other two, a mixture of one-fifth charcoal and four-fifths wood.

The competition showed that with an engine designed for gasoline, there was a loss of power when gas was substituted. Taking the power of the gasoline engine as 1, with producer gas the power would drop to between 0.50 and 0.73 if the engine were not altered in any way, and to between 0.70 and 0.90 if changes were made in the compression, size of valves or areas of valve-openings and gas passages. It has been considered that the results of the trials show that there is an economy of 25 per cent in the use of gas compared with gasoline.

The Thornycroft powerplant differs essentially from others on the market in one or two features. The steam is produced in a separate generator, which is not attached to the producer, by heat from the engine exhaust on the principle that the heat of the exhaust gases must necessarily be proportionate to the power developed, and thus the amount of steam can be made proportionate to the working. This generator can be detached from the engine and entirely taken to pieces for cleaning, which is a point of very great importance. The Thornycroft powerplant is intended primarily for use with the chassis of the British War Office pattern, known as the J type, and is designed as an integral part of the machine. Consequently, every part is easily accessible, in no way interfering with the fitting of an ordinary cab or body and detracting but little from the appearance of the vehicle. Moreover, the weight can be cut down to a minimum.

Generally speaking, the engine can be started on gas in less than 10 min. from cold, after which it is said that the car can be left standing for an indefinite period, the engine idling, with a certainty of starting away as soon as the throttle is opened, whatever the load.

No water is used for the cleaning or scrubbing of the gases, and cleaning is thereby much simplified, while the possibility of any corrosion or electrolytic action taking place is eliminated.

The Problems in Front of the Automotive Engineer

By C. F. KETTERING[1] DAYTON SECTION PAPER

MR. KETTERING mentions present practice with regard to engineering education and states that there is beginning to be a realization that such a great amount of specialization is not necessarily accomplishing the best results. He makes a plea for greater consideration by the engineer of the two fundamental factors, the economic determination of the details of design and the appeal to the final purchaser as being pleasing to him in performance, comfort and utility, because the difference between present and former automobiles is not in principle but in the way the materials used are arranged with respect to each other.

Following a brief general discussion of the transfer of energy in which he also advocates making a greater use of research methods and the data obtained in other lines of endeavor, Mr. Kettering then comments upon some of the constituents of car development, placing emphasis upon the necessity of providing an appeal to the user, and concludes his paper with a brief resume of cost considerations.

THERE is a marked similarity between the problems now confronting automotive engineers and their reaction to them, and the problems in front of the engineers of every other phase of engineering. This condition is so general that those in charge of engineering education are giving more study to the organization of their courses, and are questioning the values of the subjects taught and the methods of teaching them in a way that has not occurred for many years. This is the general condition of all engineering schools and must be considered as a reflection of the condition that all types of engineers are being called upon to meet.

At the present time, engineering education is very highly specialized. Young men intending to go into one branch of engineering take different courses from different instructors than other men in the same institution who are expecting to go into engineering work that is classified as being slightly different. There is beginning to be a realization that this large amount of specialization is not necessarily accomplishing the best results.

The undergraduate has a smattering of details applying to his own specialty, these details being usually outofdate before he is in a position of sufficient responsibility to be able to apply them; and, too frequently, he does not see the general foundation upon which he works, which is really the foundation on which all engineering work is done.

When we look at the problems that the engineer has to meet today, we find, finally, when we judge his work, that the thing which he has really done is simply a rearrangement of the material used. The difference between the automobile of today with its high degree of performance and the automobile of 1904, is not a difference in principle, but a difference in the way the materials used are arranged with respect to each other. This involves not only a reproportioning of parts, but also, to a much larger extent than we usually realize, the rearrangement of the molecules that go to make up the materials from which these parts are made. We have new materials today, and can get the same materials at a lower price which were not available to the engineer of 1904 and, as a result, we can get a new and, in our opinion, a better result by taking advantage of the situation. When the engineer begins to rearrange these materials, he has two factors of fundamental importance to keep in mind and, if he fails, it will be because he has neglected to take the proper account of one or the other. The first factor is, the economic determination of the details of the design; that is, the arrangement of the proportion of the parts in their relation to each other, so that the performance specifications are met with the least amount of material which has been subjected to a minimum cost of fabrication.

It is not enough, however, to have produced the most economical design, if that completed design does not appeal to the final inspector, the final purchaser, as being pleasing to him in performance, in comfort and in utility. A designer may accomplish some wonderful feat of performance in the design of an automobile; but if this performance does not bring any satisfaction to the owner of the vehicle, or if this performance cannot be obtained without discomfort to himself, or sometimes, what is of more importance, without pleasure to his wife, he will not buy the product and the design will be a failure.

When the engineer commences to study the possible rearrangement of material that will enable him to obtain a better or more economical result, he must first remember that materials are used in general for two purposes. The first of these is to prevent motion; and the action of the material may be considered as mainly static. A bridge or the frame of a machine obviously is of this type; the main purpose of the frame of the automobile is to prevent relative motion of the various parts. The other use of materials is to facilitate motion or, perhaps, better to guide motion and to transmit energy. The automobile, as a complete assembly, is of this type; but within the automobile, the piston is, of course, a piece of material used to transmit the motion of the expanding gases in such a way that the motion of the car itself can be obtained from it.

TRANSFER OF ENERGY

All transfers of energy involve the product of a force and a velocity. The velocity, obtained as a result of the action of the farce, is limited always by two factors: (*a*) the resistance to the overcome and (*b*) the inertia to be accelerated. After the motion has become uniform, the only factors involved are the applied force and the resistance restraining the motion due to that force.

The electrical engineer is early taught Ohm's Law. It is that the current is equal to the voltage divided by the resistance and we write it thus, $I = E/R$. We look at that as a statement of a law applied to the electrical engineer only and too often forget that it is a law of universal application and might simply be stated, velocity

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equals force applied, divided by resistance to motion. This is the equation that determines the maximum speed obtainable from an automobile and is just as true on the road or on the racetrack, as is Ohm's Law in the electrical circuit.

The question of how fast energy can be transferred from one point to another, or from one body, say, a mass of flaming gases in an automobile cylinder, to the energy dissipated in the air and on the road by the speeding car, depends entirely upon the properties of the materials involved and their arrangement. These properties are inertia, elasticity, and resistance to relative motion, which we call friction.

Too often the engineer does not understand the meaning of inertia. Very early when studying physics, he had considerable trouble with a subject called the laws of gravitation, and too often he did not realize when studying these that what he was really studying was the laws of accelerated motion, in the special case where the accelerating force is the attraction of the earth; and that the really important things he could get from this had only incidentally to do with gravitation and with weight; that the important things were force, inertia and the resultant motion coming from the application of the force to the inertia under different conditions as to frictional restraint. It would not have taken us nearly as long to work from the design of 1904 to a design similar to that of 1922, if the fundamental importance of these factors in every energy translation had been realized.

It will be worthwhile to digress for a moment and point out how scientists working in one line of research have had to take account of the methods and the information available in other lines to accomplish the results that are being accomplished today. The general public thinks of the Mount Wilson Observatory as a place where men spend their time watching the stars with great telescopes. There is, of course, today a great amount of cataloging and study of motions going on, which was the only work of the early astronomer; but by far the greater part of the work of an institution of that kind is along lines that the early astronomer never dreamed of and would have considered ridiculous had anyone been able to tell him of them.

What they are doing at Mount Wilson today is studying the spectrum of the elements under all conditions of pressure and motion; under all conditions as to magnetic and electrostatic fields that they are able to devise. The motion of the stars is determined by photographs, taken from time to time, which can be compared at the convenience of the astronomer. The work of mapping the heavens has been greatly simplified by the use of photography, and the energies of the astronomer released to permit attacking problems involving the structure of matter. The solution of these problems will have a very great effect in everyday life. The methods that are being used are methods developed by the physicist and the physical chemist, and are very far removed from the methods that were used by the astronomer of a few decades ago. The automotive engineer must follow the lead of the astronomer and borrow other methods to supplement those he has used up to this time, if he is to progress as he must to meet the new conditions.

Returning to our question of falling bodies, we find that the so-called laws of falling bodies are really laws of the transfer of energy, involving the application of force to inertia. The potential energy that a body formerly had by reason of its position is transferred to the inertia energy it has by reason of its motion. If the body falls through a dense fluid like water, the velocity is much less than if it falls through the air, and this velocity soon becomes uniform, the velocity being determined by the formula $V = F/R$, just as in the case of the electric circuit, where $I = E/R$. If the body is restrained by an elastic element such as a spring, we have energy stored in this spring, just as we have energy stored in an electric condenser. If the energy is applied or released abruptly, we get an oscillation in either case. The rebound that throws you into the top of the car, is the same kind of a thing as the electric rebound that sends out the wireless signal; and the same formula can be used in computing both phenomena, if we use the proper constants for the different cases.

When we make a tensile-strength test we are merely measuring the molecular attraction between molecules in the metal. The coefficient of elasticity, the tensile-strength, the density, are only bits of evidence as to the structure of the material and the nature of the forces holding the parts together. We cannot yet interpret all of this evidence, but we must learn to do so, so that not only our sciences may progress, but also our own automotive progress continue.

Car-Development Constituents

The car of 1922 differs from the car of 1909 only in changes in the proportions of the parts, and in the relationship of the forces and materials. The car of 1930 will differ from the car of today only in the same way. It is possible to point out some of the problems that will be solved completely, or in part, in the near future. These problems will be solved by the measurement of these relationships, and the determining of better proportions.

Take spring-suspensions and shock-absorbers, for example. When the wheel hits a bump in the road, the wheel starts to oscillate under the body and, if the oscillation persists, the body will take up motion as a result. Strike a bell, and the bell begins to vibrate, just as the wheel vibrates; and if we touch our finger to the bell the vibration stops. All the shock-absorber can do is the same thing, and the problem is to apply this resistance without interfering with the useful operation of the spring. If we have too much resistance in our wireless apparatus, it will not oscillate. In this case the resistance is injurious, as it interferes with the normal operation, but its action is exactly the same as the action of the shock-absorber.

Just recently we were shown a model of car made by a certain manufacturer; the engineer of this factory stated that this particular model was absolutely the best-riding car they had ever produced, and they could not tell why. The reason they could not tell was that they had not measured the fundamental relations that contribute to good riding.

What we need is more analysis of fundamentals and less invention. If we do not have a new inventor with a spring wheel come to the laboratory every week, we think something is wrong. If these inventors would analyze the situation, they would see that they were under an enormous handicap so far as competing with the pneumatic tire is concerned. When a pneumatic tire runs over a small bump in the road, the rim of the wheel and car proceed without being diverted from their path and without disturbance to the passenger. It is only necessary to move a small amount of rubber out of the path it would have followed, the energy involved is small, and the resulting shock is very small. When the solid rim of the spring wheel strikes the same bump, it is forced to deflect, and therefore accelerate a very much larger mass, resulting in the transfer of a much larger amount of

energy, and naturally a much greater shock to the vehicle and its passengers.

We shall see some real progress in riding qualities in the near future. Today the automobile salesman will tell you how the engine in his car will go 1000 or 1500 miles on a gallon of oil. When you buy the car you get an instruction book which tells you to drain the oil every 300 miles. You expect to discount what an automobile salesman tells you a little, but you do not expect to have to use so high a discount factor as this.

The gasoline we have today with an end-point of 440 deg. fahr. is very different from the fuel of the early automotive days. It really is not so much worse as a fuel than former gasolines, as it is different; and we must learn to use it efficiently. We can do this. We know enough about fuels, and methods of using them, to get very much better results out of the present-day fuels than were obtained with the fuels of 10 years ago. It is only necessary to build our knowledge into our new engines. Our problem is to do this in the simplest and most direct manner, and in such a way that the user will be least conscious of the change.

This brings us back to a point mentioned before, to which we must always give attention; the all-important question of providing an appeal to the user in our product. It is of no profit to an engineer and a manufacturer to produce a car that has more speed than any other, or will do more stunts than its competitors, if the public does not see at once a real gain in utility, in comfort to the user. If this gain is not evident, the public will go on buying cars of the older design, and be entirely right in doing so. It is sometimes better to go forward slowly, than to try to go too fast. Some of you may remember that the first single-cylinder cars had whipsockets on a dash that was no longer needed, for the arrangement of the car was very different than now. The public was used to seeing whipsockets on dashes, and bought vehicles so equipped with less hesitation than other types. This may be very extreme, but it illustrates a very human reaction toward change, and we must take account of this as we meet the problems of comfort and good riding, and the efficient use of oil and fuel.

Cost Considerations

I will close by pointing out once more that these problems must be solved at a minimum cost. All engineers realize that costs must be kept down and all engineers believe that they are taking account of costs. As a matter of fact, they are not giving enough thought to the most important cost of all, the so-called overhead cost. Many engineers are not studying it at all, not realizing that they are determining what the minimum overhead-cost of manufacturing is to be when they lay out their designs. Costs are divided into three factors; labor, material and overhead. If our designs contain details requiring special tooling or elaborate systems of inspection to secure the results desired, we have specified a high manufacturing overhead on our drawings, and our total costs will be high in spite of any effort we can make to reduce materials or the direct labor. When a piece is laid-out on the drawing-board, that is the time the method of manufacture should be worked out. For a given production-schedule, the tools required, the floor-space and the materials in process of manufacture can be determined, and what our capital investment is will be known. The losses in manufacture and the supervisory force are fixed by the number of operations and their difficulty. It may seem that the engineer is reaching out a little beyond his own field in attempting to analyze the overhead but, so long as the cost-accountants figure overhead as a fixed percentage, someone will have to get into the elements of this cost, if we are ever to reduce it; and we must reduce it. It is not reasonable to hold the engineer responsible for the overhead actually existing in many factories, but he does specify what the minimum overhead can be for a given production schedule, and he must accept the responsibility involved.

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Automobile Finishing-Varnish

By L. Valentine Pulsifer[1] Annual Meeting Paper

As the success or failure of the finish of an automobile depends largely on the finishing-varnish, a plea is made for more scientific analyses of the problems of automobile finishing and more care in selecting and applying a suitable varnish. The qualities to be desired in a finishing-varnish are divided into two classes: the shop qualities and the service qualities. The shop qualities include color, body or viscosity, working, flowing, setting, hardening, fulness and the safety of working. The service qualities, or those that enable the varnish to withstand the various conditions of use, include resistance to break-down under the chemical action of the actinic rays of sunlight, to the destructive action of moisture and the alkalis in mud and soap, to expansion and contraction, to vibration and to abrasion. The three most important factors in estimating the service-giving qualities of varnish are said to be elasticity, moisture-resistance and the film factor. Each of the various terms mentioned is carefully defined, analyzed and explained. Among the constants that must be determined, either for checking uniformity or for use as bases of tests, are the non-volatile content, which is the starting-point in the test for elasticity and in computing the film factor or thickness of the dried film; the ash, the flash-point, and the acid number. An empirical formula is derived by which the thickness of the dried film can be estimated with accuracy; and in a series of appendixes the details are given concerning tests for the drying, the safety of working, non-volatile content, ash, flash-point and acid number; the Kauri reduction test; and tests for moisture and alkali resistance.

Last year before the Society of Automotive Engineers and the Automobile Body Builders Association I discussed the Manufacture and Application of Automobile Varnishes and Paints[2] and Some Problems in Automobile Finishing.[3] These two talks covered the basic principles of the manufacture of varnishes and paints and the fundamental engineering problems involved in the building of a successful automobile-finish. At the conclusion of the second of these talks I made the following statement:

> I hope the points and problems I have discussed will arouse a greater desire to study this particular field of automotive engineering, both on the part of the automotive industry and on the part of the paint and varnish industry. A scientific study of the problems involved, by as many qualified groups as possible, will yield ample dividends of satisfaction to all concerned.

This plea for cooperative effort along the lines suggested has roused considerable interest; and an attempt is being made to substitute knowledge for the hit-or-miss methods that in the past have been all too prevalent. With a desire to assist in establishing automobile finishing on sound engineering principles I shall take up the subject of automobile finishing-varnish in greater detail than time permitted in the two talks mentioned above.

Automobile Finishing-Varnish

Assuming that the undercoats of an automobile-painting system have been properly selected and applied and that the finished car will receive intelligent and proper care from its prospective owner, the success or failure of the finish will depend on the finishing-varnish.

The fact that the finish of many automobiles, even those of the highest grade, does not last so well as did the finish on the fine carriages of a generation or two ago, has led some persons to believe that the general quality of finishing-varnishes has deteriorated. Such a conclusion is the reverse of the truth, for it is possible today to obtain varnishes of far greater durability and much greater resistance to abuse than could be obtained from 25 to 50 years ago. The conclusion, however, is not unnatural if one does not consider that the finish on an automobile, even with normal use, undergoes a treatment that the carriage finish would never have received, and that thousands of automobiles are finished with varnishes that in the days of the carriage would hardly have been used even for quick cheap repair work. When speed, cheapness and the need of covering-up defects in the rubbing-coats are the compelling motives, finishing-varnishes sometimes are used that hardly exceed in durability a good elastic rubbing-varnish; but this practice usually is due to a lack of knowledge of the consequences involved. Even when these are not the determining factors, a finishing-varnish frequently is used that represents perhaps a scant 25 per cent of the maximum durability obtainable. Here, again, a lack of knowledge is involved, which, if it existed in the whole field of automotive engineering, would doom the automobile to hopeless failure. The use of a finishing-varnish with rubbing-varnish elasticity is about on a par with the use of cast iron for crankshafts, or white pine for spokes.

It is my purpose to describe the qualities required of a successful automobile finishing-varnish and the methods of determining whether these qualities exist in any given varnish.

Qualities to Be Desired

The most desirable qualities of an automobile finishing-varnish must be determined by the automotive and varnish industries jointly, for each understands best the limitations under which it has to operate. Compromises must be made between what is desirable and what is possible. This is true of varnish as well as of steel, leather and other automotive materials; and it is only when both parties desire to produce the best that the ultimate in automobile finishing can be obtained. The selection of the finishing-varnish best suited to a given car, a given production-schedule and a given method of application must be made after a combined study by the varnish manufacturer and the automobile builder.

To do this without great waste of both time and material, rapid methods of estimating must be used and the results of such estimating must be intelligently interpreted. The important qualities for convenience may be divided into two classes: (*a*) shop qualities and (*b*) service qualities; these will be taken up separately.

Shop Qualities

Shop qualities include color, body or viscosity, working, flowing, setting, hardening, fullness and the safety

[1] Chief chemist, Valentine & Co., New York City.

[2] See The Journal, January, 1922, p. 12.

[3] See *Automotive Manufacturer*, August, 1922, p. 26.

of working. To estimate them satisfactorily is easier than to estimate the service-qualities of an unknown varnish; and some of the qualities such as working, flowing, setting, hardening and the fullness of finish can be determined best in the shop itself under average shop conditions. By "working" is meant the possession of the proper working-properties for the method of application it is desired to use, whether by brush or by flow-coat. By "flowing" is meant the ability to flow-out properly without runs, sags, curtains or brush-marks, and yet to retain a suitable body of varnish on the finished surface. By "setting" is meant the ability to set slowly enough to allow the entire surface to become smooth and yet fast enough to prevent too long a flow, which sometimes permits the varnish to run over a molding or from an upper panel to a lower one. The "hardening" and "fullness" are estimated most easily on a body run through on the regular production-schedule, but the quality of hardness can also be determined satisfactorily by a laboratory test that duplicates shop drying conditions. The "safety of working," that is, the ability of the varnish to act properly under adverse shop conditions, is hardly an important factor in a modern automobile-plant. For drying and safety-of-working tests, see Appendix 1.

The two other shop qualities enumerated above, color and body or viscosity, are best estimated by using the Gardner-Holdt viscometer, an instrument developed in the research laboratory maintained in the City of Washington by the National Varnish Manufacturers Association. This is a bubble viscometer consisting of a series of 20 tubes standardized as to internal diameter and filled with standardized compounded petroleum oils. These oils vary in color and in viscosity; and each tube, marked *A*, *B*, *C*, *D*, and the like, contains an oil ½ bubble more heavily bodied than the preceding, and a shade darker in color. In 10-tube sets the variation between one tube and the next is one full bubble, which provides close enough gradation in both viscosity and color for the automotive laboratory. It takes a difference of more than one full bubble before the difference in viscosity will become noticeable under shop conditions. Each viscometer is provided with several extra standardized tubes in which are placed the varnishes to be tested.

The color determination should be made by placing the tube filled with the varnish and the different tubes of the viscometer, one at a time, close together, and looking through them by transmitted light, preferably daylight. If the color of the varnish matches the tube *D*, or is between *C* and *D*, its color can be recorded as "*D*" or "*C-D*." Other things being equal, the paler a varnish is the better it is, especially when the preservation of the purity of tone of light or brilliant colors is desired. It is convenient to use these same standard tubes of the Gardner-Holdt viscometer, which in color cover the range of clear automobile varnishes, for reading the color of a given varnish.

The viscosity is determined by holding the tube containing the varnish and one of the tubes of the viscometer exactly parallel in a vertical position and then rapidly reversing the tubes. When the tube of the set is found in which the rate of travel of the bubble is exactly the same as in the tube of varnish, the viscosity of the varnish is read as, say, F-1.40 poises, the poises, or measure of absolute viscosity, having been determined for the different tubes of the viscometer. This test should be made carefully to avoid any differences of temperature in the tubes of varnish or oil and the temperature preferably should not vary more than 1 deg. above or below 77 deg. fahr.

As different methods of application and varying shop conditions and temperatures demand varnishes of different viscosities, the method and the apparatus outlined above are the best and most practical that have been devised.

The statement that some shop qualities can best be tested in the shop itself should be qualified by saying that if the tests are so made they should be under adequate technical supervision and carefully checked by the engineering department to assure standardized conditions and authentic records. The subject of service qualities falls almost entirely within the laboratory.

Service Qualities

By service qualities are meant the qualities that determine the ability of a varnish to give service, that is, to stand up under the numerous varying conditions of automobile use. Before deciding what these qualities are it is necessary to understand the causes of the final break-down that comes eventually to all varnishes. The most important is the chemical action of the sun's rays, that is, the actinic rays of sunlight. These slowly break-down the vegetable compounds in the dried film of varnish and promote progressive oxidation or "rotting," thereby causing a gradual loss of elasticity. When the elasticity of the film is reduced below that which is necessary to withstand the expansion and contraction of the surface underneath, and the vibration, small cracks appear and the final break-down approaches.

The importance of the chemical rays in sunlight, as the prime factor in the destruction of finishing-varnish, is evident when one observes the perishing of decks and cowls long before the more nearly vertical panels of the body. The surface that gets the sun's rays most nearly perpendicularly will perish first; in fact, on test exposure-panels, a panel exposed at an angle of 45 deg. from the vertical facing south will perish in about one-half the time of an exactly similar panel exposed vertically, and a panel exposed vertically facing north will last approximately twice as long as the one exposed vertically facing south. The effect of the angle of exposure is not always understood, as is shown by the fact that the hoods of automobiles are frequently finished with a varnish less durable than that used on the bodies.

On the top panels of most hoods the angle is about the worst possible, the expansion and contraction owing to changes of temperature being more severe and the vibration very much worse than anywhere else on the car.

A factor contributing largely to this gradual break-down is moisture or moisture plus alkali, in the form of soap or of mud. The presence of moisture in the varnish-film aids the chemical action of the sun's rays in the final break-down of the varnish. The destructive action of alkalis such as mud and soap in connection with varnish is, of course, well known. Abrasion also is a contributing cause of the perishing of the coat of finishing-varnish, for the scratches in the surface caused by careless washing or the use of dry washes or polishes when dust or dried mud is on the car, develop eventually into cracks.

To sum up the causes of the deterioration and eventual destruction of the finishing-varnish, we have, first and most important, the chemical action of sunlight, aided by moisture, alkalis, expansion and contraction, vibration and abrasion.

It is clear that to postpone this final break-down a varnish should possess as great an initial elasticity, as high a resistance to the destructive chemical effect of moisture and as thick a dried film as are permitted by the method of application and the time-schedule.

Estimation of Service Qualities

After the automotive engineer has decided on the type of finishing-varnish he wishes to use, he should have the means at his disposal first for finding and then for check-testing the deliveries of the material selected. A lack of such laboratory means would spell chaos if it extended over the whole list of automotive materials, yet many otherwise well organized plants lack the laboratory means for testing and estimating varnish.

The varnish industry as a whole is perhaps as responsible for this lack as is the automotive industry, although, as already mentioned, the National Varnish Manufacturers' Association maintains in the City of Washington a research laboratory that is devoted to the production problems involved in varnish manufacture.

The research laboratory of the company that I represent has striven earnestly to perfect and to standardize the methods of estimating the qualities desirable in a varnish; and some of the methods have been adopted by Government laboratories and by the laboratories of certain railroad, automotive and industrial units. The three most important factors in estimating the service-giving qualities of varnish are elasticity, moisture-resistance and the film factor.

In beginning the examination of a varnish it is useful to determine certain constants that are used in checking uniformity or as bases for other tests. The most important is the non-volatile content, as this is used as a starting-point in the test of elasticity and as one of the necessary elements in computing the film factor or the approximate thickness of the dried film. Next, chiefly for checking purposes, is the ash, which, if it does not exceed 1 per cent, is not important; the flash-point, which, to assure proper working qualities and low fire-hazard, should not be below 105 deg. fahr.; and the acid number, which is unimportant except as a checking figure. The proper methods for determining non-volatile content, ash, flash-point and acid number will be found in Appendix 2.

Elasticity

Elasticity is the most important factor in determining the service-durability of a finishing-varnish and, other things being equal, the comparative durability of a series of varnishes follows exactly their comparative elasticity. This factor can be estimated in the laboratory by the so-called Kauri reduction-test for elasticity. This test was developed by me several years ago after a series of experiments made with a view to perfecting a laboratory method for rapidly and accurately estimating the elasticity factor. It was adopted by the Government during the war for airplane use and has since appeared in numerous specifications, replacing the former slow and unsatisfactory panel-exposure test.

The test is based on the fact that, if two varnishes are equal in other respects, their elasticity will vary with the gum-oil ratio; in other words, the higher the fixed-oil content in terms of the gum content, the greater will be the elasticity. In making the test various amounts of a standardized gum in solution in redistilled spirits of turpentine are added to the varnish. The varnish is then put through a forced drying process on a standardized tin-plate and the plate thus coated is bent rapidly over a rod of fixed diameter. The amount of gum solution necessary to bring the varnish to the cracking-point under this procedure forms a measure of its elasticity, that is, the greater the amount of gum solution necessary, the higher is the elasticity. The elasticity factor is read and recorded in terms of the percentage of gum solution that it is necessary to add to the varnish to bring it to the cracking-point, say, 20, 60 or 120, as the case may be. This test determines only the elasticity factor of the varnish and not the original gum-oil ratio, for the kinds of gum and oil used, the kind and amount of drier and the heat-treatment of the varnish during its manufacture all influence the elasticity factor of the finished product. The fact that it is possible to have two varnishes of the same gum-oil ratio that will differ in elasticity by more than 100 per cent shows why it is easier to estimate the quality of a varnish by physical tests than by analyses of the components, even were it possible to make such analyses with accuracy.

The range of percentage of the gum solution that exterior varnishes will stand on the Kauri reduction-test is from 0 to 200, the 0 per cent being on the border-line between the most elastic of the rubbing-varnishes and the least elastic of the finishing-varnishes, and the 200 per cent representing the maximum elasticity of a fixed vegetable-oil varnish that contains no gum whatever. Full directions for determining the elasticity factor of a finishing-varnish are given in Appendix 3.

Resistance to Moisture and Alkalis

Varnish is affected by moisture in two ways, physically and chemically, and by alkalis, for both the gums, or resins, and the vegetable oils used in making it, are saponifiable in varying degree. The degree in which varnish is affected by moisture and alkalis can be varied within very wide limits, by both the composition and the heat-treatment of the varnish during its manufacture. For purposes of comparison, the rate at which different varnishes absorb moisture can be determined with sufficient accuracy by observing the rate at which they turn white or opaque when they are immersed in either hot or cold water. Susceptibility to chemical injury by water can be measured by ascertaining how long they must be subjected to the action of water, either hot or cold, before they become permanently whitened, for permanent whitening indicates a chemical change in the structure of the varnish. Comparative resistance to the destructive action of alkalis can be determined by immersing the varnishes in a standard soap-solution for short periods of time, the action of the alkalis on the gums and oils being indicated by loss of lustre and whitening.

It is necessary to make these determinations because they have an important bearing on the service-giving qualities of the varnish. In this connection it should be remembered that resistance to both water and alkalis is of greater importance in the case of a varnish that is to be used on the hood, fenders, wheels and axles than of a varnish that is to be used on the body. In a body-varnish, a resistance to these two elements as high as can be attained without sacrificing the necessary elasticity and freedom of working and flowing is desirable, and, other things being equal, the body-varnish that resists moisture and alkalis most effectually is the best choice. A description of the methods for making tests for resistance to moisture and alkalis is given in Appendix 4.

Film Factor

By the film factor is meant the normal thickness of the dried film. The durability of a finishing-varnish varies almost directly with the thickness of the film, provided, of course, the film is not too thick to harden through properly. In the case of undercoats a thick film is to be avoided, but in the final coat the thicker the better, provided, as said before, it is not too thick to dry and to harden properly. Actual measurements of film thickness involve considerable trouble and are difficult to make

accurately. But a sufficiently accurate calculation can be made from data used in the other tests. These data are the viscosity of the varnish, in poises, and the non-volatile content, in per cent. By using the following equation the normal thickness of the dried film can be estimated with surprising accuracy, as has been demonstrated in our laboratory by a large number of check-tests.

$$F = \tfrac{1}{2}(10V + N)$$

where

F = the film factor in microns
N = the percentage of the non-volatile content
V = the viscosity in poises

For example, a body-finishing varnish having a viscosity of 1.40 poises and a non-volatile content of 64 per cent will figure as follows:

$$F = \tfrac{1}{2}[(10 \times 1.40) + 64]$$
$$= 39$$

A 2-ft.-square glass panel finished with a normal coat of this particular varnish and measured with the aid of a powerful microscope gave average readings of exactly 39 microns.

This determination of the film factor completes the data on which the service-durability of the varnish is to be estimated; next comes the assembling and the intelligent appraising of the data. Perhaps the easiest way to explain the methods used is to take four finishing-varnishes, *A*, *B*, *C* and *D*, which are in use in quantity production and on which the elasticity, the moisture resistance and the film factor have been determined, and to rate them according to their estimated service-durability. By "durability" is meant the ultimate protection afforded the paint structure on an automobile body under actual service conditions, and not the first signs of checking or cracking looked for on a laboratory exposure-panel. The four varnishes selected represent, approximately, the minimum and the maximum limits in color, viscosity, non-volatile content, film factor, moisture resistance, elasticity and service-durability of the varnishes in current use for finishing automobile bodies.

The successful comparative rating of varnishes on these data requires careful consideration coupled with experience in varnish testing and research. This is especially true of the resistance to moisture and alkalis. If, however, on the basis of the water and alkali tests outlined above and detailed in Appendix 4, the varnishes are given moisture-resistance factors, a reasonable rating will result for average service conditions. A varnish that stands these tests poorly will receive a 0 rating; fair, 10; good, 20; and very good, 50.

To determine the comparative service-durability ratings of the four varnishes, we add together the figures obtained for each in our tests for elasticity, moisture resistance and film factor. This gives the results presented in Table 1.

TABLE 1—SERVICE-DURABILITY RATINGS OF FOUR FINISHING-VARNISHES

Finishing-Varnishes	*A*	*B*	*C*	*D*
Elasticity Factor	10	60	90	160
Moisture Resistance	0	10	20	50
Film Factor	35	47	52	36
Service-Durability Rating	45	117	162	246

The film factors for these different varnishes were arrived at by calculation in the formula given above and were based on the data contained in Table 2.

Based on the above rating, varnish *B* would have a service-durability approximately 150 per cent greater than that of varnish *A*; varnish *C* would exceed the service-durability of *A* by about 260 per cent and of *B* by about 40 per cent; and varnish *C* would show a service-durability that is more than 400 per cent greater than that of *A*, more than 100 per cent greater than that of *B* and approximately 50 per cent greater than that of *C*. Expressed in length of life under service conditions, presupposing that the undercoats had been properly applied and that the finish of the car had received intelligent care, varnish *A* would give about 4 months of service, varnish *B* would last about 1 year, varnish *C* would survive almost 1½ years of service and varnish *D* would last 2 years, before needing renewal.

TABLE 2—DATA ON WHICH THE FILM-FACTOR VALUES IN TABLE 1 WERE BASED

Finishing-Varnishes	Viscosity, Poises	Non-Volatile Content, Per Cent
A	1.25	58.75
B	3.20	63.75
C	3.20	73.00
D	1.65	56.00

The fact that varnishes vary in durability by as much as 500 per cent may prove surprising. The facts, methods and conclusions I have presented form an outline that the automotive engineer can expand or condense according to his needs.

The information given and the suggestions made may be completely and concisely put into three words: *Know Your Varnish!*

APPENDIX 1

Drying and Safety of Working Tests

To test the drying and hardening of the varnish in the laboratory the following standard method will be found satisfactory: Pour the varnish to be tested on a bright tin-panel, 3 x 5 in. in size, that has been carefully cleaned with benzol, and stand the panel in a nearly vertical position in a well ventilated room kept at the same temperature as the shop in which the varnish is to be used, say 70 to 80 deg. fahr. The panel must not be placed in the direct rays of the sun. From time to time test the film by touching it lightly with the finger at points not less than 1 in. from the edges of the film. The varnish is considered to have dried to touch when gentle pressure shows a tacky condition but no varnish adheres to the finger. The varnish is considered to have dried hard when the pressure that can be exerted between thumb and finger does not move the film or leave a permanent mark after the spot has been lightly polished.

An over-sensitiveness to adverse shop conditions can be determined by the following simple test: Pour an excess of the varnish on a small 3x5-in. tin-panel that has been carefully cleaned with benzol, and immediately place the panel in the direct draft of a small 8 to 10-in. electric-fan, running at full speed. The panel should be placed in a nearly vertical position approximately 2 ft. from the fan and at an angle of 45 deg. to the line of the air-current. Allow the panel to remain in this position 5 hr.; remove and allow the varnish to harden overnight. The varnish should show no dulling, enameling, crow-footing or frosting. This test should be made under shop conditions that are normal as to ventilation and temperature.

APPENDIX 2

Laboratory Determinations

NON-VOLATILE CONTENT

Place a portion of the sample in a stoppered bottle or a weighing pipette. Weigh the container and the sample. Transfer about 1.5 grams of the sample to a weighed

flat-bottomed metal dish about 8 cm. in diameter, a friction-top can plug. Weigh the container again and by the difference calculate the exact weight of the portion of the sample transferred to the weighed dish. Heat the dish and contents in an oven maintained at 105 to 110 deg. cent. (221 to 230 deg. fahr.) for 3 hr. Cool and weigh. From the weight of the residue left in the dish and the weight of the sample taken, calculate the percentage of non-volatile residue.

ASH

Determine the ash in 10 grams by ignition, in a quartz or a porcelain dish, carrying out the determination at a low heat, preferably in the muffle. If a muffle is not available, the thinner should first be driven off on a hot plate or in an air-bath before direct ignition. Particular care must be taken not to over-hasten the procedure during ignition, as a reduction of the lead drier salts would invariably follow.

FLASH-POINT

The open-cup method may be used, as described in the tentative standard of the American Society for Testing Materials, No. D92-21T, or else the closed-cup method, according to Standard No. D56-21 of the American Society for Testing Materials.

ACID NUMBER

Weigh 10 grams of varnish in a 200-cc. Erlenmeyer flask, add 50 cc. of neutral alcohol, connect with a reflux air-condenser and heat in a steam-bath for ½ hr. Remove from the bath, cool, add phenolphthalein and titrate the free acid with fifth-normal sodium hydroxide. Calculate the acid number, that is, the milligrams of potassium hydroxide to 1 gram of varnish.

APPENDIX 3

The Kauri Reduction Test

Preparation of the "Run" Kauri.—Arrange a distillation-flask, a water-cooled condenser and a tared receiver on a balance. Place in the flask about one-third of its volumetric capacity of clear, bright hard pieces of Kauri gum, broken to pea-size. Carefully melt and distill, until 25 per cent by weight of the gum is collected in the tared receiver. At the end of the distillation, the thermometer in the distillation-flask, with the bulb at the line of the discharging-point of the flask, should register about 600 deg. fahr. Pour the residue into a clean pan and, when cold, break into small pieces

Preparation of Standard "Run" Kauri Solution.—In a carefully tared beaker place a quantity of the small broken pieces of the "run" Kauri with twice its weight of freshly redistilled spirits of turpentine, using only the portion that distills over between 308 and 338 deg. fahr. Dissolve by heating to a temperature of about 300 deg. fahr. and bring back to correct weight when cold by the addition of the amount of redistilled spirits of turpentine necessary to replace the loss by evaporation during the dissolving of the gum

Reduction of the Varnish.—First, carefully determine the non-volatile content of the varnish under test according to the method outlined in Appendix 2. Then take 100 grams of the varnish and add to it an amount of the standard run Kauri solution equivalent to 50 per cent by weight of the non-volatile matter in the varnish. Mix the varnish and the solution thoroughly. For the sake of simplicity the amount of varnish to be used is given as 100 grams, whereas 25 grams is sufficient, and taking the smaller amount will conserve the supply both of the sample and of the standard run Kauri solution

Flow.—Place a coat of the varnish thus reduced on a standard 3 x 5-in. tinned panel, carefully cleaned with benzol, and allow it to dry in a nearly vertical position at room temperature for 1 hr. Next place the panel in a horizontal position in a properly ventilated oven and bake it for 5 hr. at from 96 to 100 deg. cent. (205 to 212 deg. fahr.). Remove the panel and allow it to cool at room temperature, preferably 75 deg. fahr., for 1 hr.

Elasticity Test.—Place the panel with the varnish side uppermost over a 2-mm. (⅛-in.) rod, held firmly by suitable supports at a point equally distant from the top and the bottom edges of the panel, and bend the panel double rapidly. The varnish should show no cracking at the point of bending. For accurate results the bending of the panel should always be done at 75 deg. fahr., for lowering the temperature will decrease the percentage of reduction that the varnish will stand without cracking; while raising the temperature will increase the reduction.

If the varnish, when tested as outlined in the preceding paragraph, cracks with the 50-per cent reduction specified, make up additional samples reduced with lesser amounts of the standard Kauri solution until a point is reached where it does not crack. If, on the other hand, the varnish does not crack when reduced 50 per cent, prepare other samples reduced with larger amounts of the standard Kauri solution, until the cracking-point on subsequent baking and bending is determined

The panels required for this test are of bright tin, 3 x 5 in., and approximately 0.278 to 0.297 mm. (0.0109 to 0.0117 in.) thick. The weight of the metal per standard base box, 112 sheets 14 x 20-in., Nos. 31 and 30½, United States standard plate gage, is 90 to 100 lb. They should be thoroughly cleaned with benzol immediately before using

This test, carefully followed out, will enable the varnish-testing laboratory to rate the comparative elasticity of a number of unknown varnishes within 2 working days and with greater accuracy than is possible by the panel weather-exposure test, which requires several months. It can also be used for checking the elasticity of deliveries against previous standards. As the elasticity of a varnish is the chief factor in determining its durability, the value of this test, in both time-saving and accuracy, over that of former methods is very great.

APPENDIX 4

Moisture and Alkali-Resistance Tests

For determining the moisture and alkali resistance of a varnish the following method will be found simple and efficient. For the tests use bright tin-panels, 3x5 in., carefully cleaned with benzol before using. The same panels specified for the Kauri reduction test may be used.

Pour the varnish on three of the panels, place them in a well ventilated room at shop temperature, preferably from 70 to 80 deg. fahr., out of direct sunlight, and allow them to dry for 48 hr. Place one of the panels in a beaker containing about 2½ in. of distilled water, immersing the end that was uppermost during the drying period, and leave it in the water at room temperature for 18 hr. Remove it and allow it to dry at room temperature for 2 hr. At the end of this period the varnish should show no permanent whitening and not more than a slight dulling. Place the second panel in a beaker containing about 2½ in. of boiling distilled water, immersing the end that

was uppermost during the drying period, and allow it to remain in the boiling water for 15 min. Remove it and allow it to dry at room temperature for 2 hr. At the end of this period the varnish should show no permanent whitening and not more than a slight dulling. If the finishing-varnish does not pass these tests it can be rated as poor on moisture-resistance. In getting the comparative moisture-resistance of a number of varnishes it may be necessary to repeat the tests one or more times. An automobile finishing-varnish need not have the high moisture-resistance demanded in a spar-varnish, but, other features being equal, the finishing-varnish showing the highest degree of moisture-resistance will remain bright and afford protection for the longest time under the conditions of automobile use.

The third panel mentioned above may be used for testing the resistance to alkalis; in this test a standard soap-solution consisting of ½ oz. of Ivory soap dissolved in 1 gal. of distilled water is used. Place the panel in a beaker containing about 2½ in. of the standard soap-solution heated to and maintained at a temperature of 120 deg. fahr. Leave the panel in the hot solution 10 min., remove it and immediately rinse it thoroughly in clear cold tap water, and allow it to dry at room temperature for 1 hr. At the end of this period the varnish should show no whitening and not more than a slight dulling. A finishing-varnish failing to pass this test should be rated poor on alkali-resistance. In getting the comparative resistance to alkali of a number of varnishes this test may be repeated one or more times, using fresh soap-solution each time. While all oil-varnishes will finally be destroyed by the action of alkali, other features being equal, the finishing-varnish showing the highest degree of resistance to its action will remain bright and afford protection for the longest time under the conditions of automobile use.

A higher resistance to the action of moisture and alkali should be required of a finishing-varnish for use on hoods, fenders, wheels and underparts than is required for body use.

The Needless Hardwood Lumber Waste

By FREDERICK F. MURRAY[1]

ANNUAL MEETING PAPER *Illustrated with* DRAWINGS

THE production of hardwood lumber has not kept pace with factory development, particularly that which has taken place in the automotive field. Some persons maintain that lumber is not susceptible to "development"; but all sorts and conditions of hardwood lumber are produced from the same tree and only a part is suitable for automotive construction. Some portions of the tree are far more suitable for furniture manufacture, while others are peculiarly adapted for use in parquetry and in building trim. Development, therefore, consists of properly differentiating the various kinds of material that come from the hardwood log so that the various industries may have at their disposal boards that have been properly classified. As the present system of grading has been in effect substantially without change for over a quarter of a century, years before the automobile came into general use, the grades originally established have little bearing upon automotive utility. Revised specifications for grading that would determine the major requirements, which in themselves include the secondary requirements, would on the whole obviate much of the waste that occurs under the present method.

Designating the grade in terms of the cut-up value would provide a real and much needed simplification and avoid many existing inconsistencies. It would tend to discourage manipulation, the substitution of inferior grades and other irregular practices, that though reasonably infrequent are detrimental to the industry. Hardwood standardization is a component part of a general movement that involves all species of lumber, both softwood and hardwood, that is crystallizing in an organization recently instituted in the city of Washington and known as the Central Committee on Lumber Standards. This committee is endeavoring to produce a new and scientific grading-schedule that by common accord will establish a lasting instrument to be entitled the American Lumber Standards.

THE increasing difficulty experienced by the automotive engineer in securing the kinds, qualities and quantities of hardwood lumber necessary for the automotive body is a pressing subject of complaint. It has been asserted that the woods of today are not as good as they were 10 years ago, and that with the excessive proportion of waste compared with the original shipment of lumber the scientific development of factory practice in increasing production, cutting costs and eliminating unnecessary waste has become a difficult task.

The very apparent uncertainty as to the yield that a shipment of hardwood lumber will give in terms of the fabricated parts, and, in general, the inability to figure lumber as an exact factory raw material, are leading the factory user to believe that hardwoods are becoming too expensive and too much of a gamble. One of the primary sources of the belief that the price of hardwoods is making their use prohibitive lies in the fact that practically all the consuming interests consider the top grades of hardwood essential for their purposes, with the result that there are not enough top-grade materials to go round.

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ESTIMATED 3 PER CENT OF BOARDS PRODUCED IS "CLEAR"

It should be noted at the outset that only about 3 per cent of the boards produced from the hardwood log is "clear," that is, free from any form of defect, such as discoloration, worm holes, knots and other characteristic blemishes that affect the appearance and strength of the boards. The top grade of hardwoods, of course, includes not only all this clear material, but also lumber with defects of a specified character and number, the number depending upon the surface measure of the board.

Although the quality of hardwood lumber is declining somewhat, an actual scarcity will not be felt acutely for many years. If corrective measures are taken to utilize present-day production better, and if increased attention is given to forest conservation and reforestation, it can logically be expected that a timber famine will be projected far into the future, many generations hence.

The use of wood in automobile body construction during the past 15 years has progressed, strange as it may seem, both forward and backward at the same time. In the individual body, its use has been on the decline, but the total consumption has gained remarkably, owing to the tremendous increase in the production of motor vehicles.

Not long ago the all-wood body, featuring the use of wide poplar, was considered to be the most desirable construction for the passenger-car body. Aluminum sheets then supplanted wood for the panels, and now in the less expensive cars we find sheet iron or sheet steel, the skeleton of wood, however, being retained. The latest development, the "all-steel" body, seeks to replace this wood framing, and is significant in that it seems to indicate that the use of hardwood has not kept pace with other developments in the automotive field; otherwise the qualities that are so desirable and found in wood alone, would not be replaced.

HARDWOODS NOT YET "DEVELOPED" TO PROVIDE FULL UTILIZATION VALUE

Briefly, the situation is that at present the consumer in the automotive as well as in other fields, is not getting the utilization value from a shipment of lumber that he is getting with shipments of other raw materials for factory use.

In this connection it should be pointed out that there has been substantially no change in the wood itself, except in its abundance. Ash is still ash and maple maple; and, in fact, the same raw material is served to the industry that has been served for many years and is obtainable under the same specifications. This is not cited as a recommendation, or to call forth approbation, but rather to carry the point that the changes have not taken place in the manufacture and distribution of hardwood

lumber that are needed to place hardwood lumber in a competitive market with substitutes. And this is a fact that "substitutes" are making capital of.

In opposition to such an argument it has been pointed out that hardwood lumber is "just lumber" and, consequently, there is no opportunity for development. This theory is erroneous in that there are all kinds and conditions of hardwood lumber. There is great opportunity for segregation to provide each consuming industry with the portion of the tree that it can use to the best advantage with a minimum of waste. Anyone familiar with the present conditions in the hardwood lumber business, from either the producer's or consumer's standpoint, readily appreciates that very little has been done to perfect the utilization of hardwoods from the tree through the factory. Without some thought the possibility of improvement in this direction is not directly apparent.

Interests of Consumer and Producer Bound Together

It should be emphasized in outlining a program to improve conditions in the hardwood business, from both the consumer's and the producer's standpoints, that any development which the producer might propose would immediately be felt by the consumer, and, likewise, that any proposition which the consumer might promulgate would affect the producer. Is it not apparent then that these two elements in the trade must go hand-in-hand in order that the interests of both may be served to their mutual advantage? On such a foundation alone can a satisfactory adjustment be made.

It has been proposed by various organizations of consumers that the producer suggest specific points for the consumer to consider and, likewise, the producer has asked the consumer to suggest points wherein an advantage might be gained. In the first case, the producer is assumed to be endeavoring to force an issue; in the second case the usual attitude is that the consumer is endeavoring to work a hardship on the producer. Unless these two elements are definitely coordinated and a thorough study is made of the possibilities and limitations of each, there can be no complete accord and elements of distrust are likely to arise.

When one endeavors to supplant a grading vehicle, originated over a quarter of a century ago and sustained practically without change since that time, a fundamental necessity is that the purpose be clearly understood from the beginning, and each phase of the development likewise, so that there shall be no dissenting voice when the ultimate end is reached.

Summarizing, it should be emphasized particularly that standardization and waste elimination in the hardwood industry from the tree through the factory is a problem common to the producer and the consumer and to these two elements alone; and that betterments cannot be instituted without a careful canvass of conditions, so that such revisions of existing practice as may be developed will be thoroughly understood as to motive and purpose, and be accepted with the knowledge that they are for the best.

Present Grades Inadequate, Obsolete and Wasteful

There is ample reason for the statement that present grades of lumber are inadequate and that the specifications long since have become obsolete. During recent decades great consuming industries have been conceived and developed, typical among which is the automotive industry that now stands second in the consumption of hardwood lumber, being surpassed only by the furniture industries. Is there not some question whether the automotive industry can be served adequately by specifications that were developed before its existence and obviously without regard to automotive utility?

A grading structure for any raw material or commodity used in factory practice can be adequate only if the purposes for which it is to be used are thoroughly considered in the grades. It is vital to the automotive industry that it should be recognized as the second largest consumer and that such provision should be made in a grading program that the parts of the hardwood tree suitable for automotive manufacture shall be made available to the industry, and not be scattered in various directions.

It is a fact, well known to every consumer, that in the existing grades there is no guarantee that any industry can realize a certain minimum yield in the factory from any one grade; and until a grading structure shall have been developed that will specify the maximum amount of waste to be allowed in fabricating to certain sizes, no consumer can gage his factory operation with any degree of certainty, and the abnormally large waste, often specified as high as 60 per cent, will continue.

Summarizing, the first inadequacy of the present grade structure with respect to the automotive industry and to every factory cut-up industry using hardwoods, is that the grades have been made without regard to the specific requirements of the factory. While admittedly each individual requirement cannot be served in an isolated way, specifying the determining requirements, which in themselves include the secondary requirements, would as a whole result in much less waste than prevails under the present method of grading.

Destructive criticism without a constructive building program is worse than useless. In brief it may be said that the answer lies in determining in a thorough and comprehensive way the cut-up requirements of the major branches of the consuming industries, basing the grade structure upon suitable composites of these utilities, and defining the specifications in terms of the cut-up requirements, thus standardizing hardwoods as a commodity with a guaranteed yield that can be determined by a straightforward yardstick method.

Reason for Obsolescence of Present Grade-Basis

The present grade structure, though wholly inadequate as a scientific grading vehicle, was developed with the most painstaking care over a period of years that marked no change in the basis plan. At the beginning the use of hardwoods was very much restricted and hardwood products went largely to the furniture and wagon trades. Today there is a very high degree of specialization in the furniture field. For example, the chair industry has clearly defined itself. There is the table industry and another specializing in bedroom suites. Progress has been made in placing the general furniture factory in the discard in favor of the shop that specializes in a limited line, but this has not contributed to a better utilization of lumber, inasmuch as the present grade-basis was in existence at the time the general furniture factory predominated. The specializing interests are now seeking a specific yield from each shipment of lumber, and it is a matter of common observation that the present grades cannot serve adequately in this particular. All the shops have departed from the "carpenter-shop" method and become production factories where a severe burden is entailed by any element that impedes straightforward progress. It is obvious that with the cut-up orders limited because of the manufacture of specialized lines, there is not the opportunity on the present

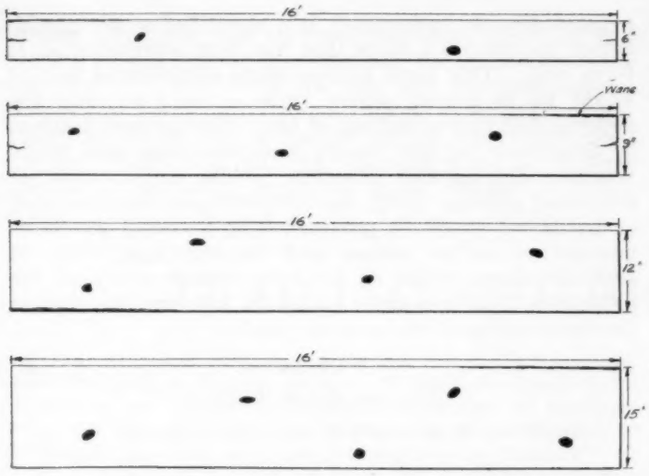

FIG. 1—PRACTICAL EXAMPLES OF BOARDS THAT ARE CLASSED AS NO. 1 COMMON HARDWOOD

grade-basis to utilize the entire board, as might be the case in carpenter-shop operation.

As a specific example, consider the top grade of hardwoods under the present grading structure, a classification known as "firsts and seconds" grade. The grade definition defines the poorest board that may properly be classed within the grade and limits it in terms of enumerated defects, such as a 1¼-in. knot.

In 16-ft. lengths a board 6 in. wide can make the top grade with two defects; a board 9 in. wide, three defects; 12 in. wide, four defects; 15 in. wide, five defects; absolutely without regard to where the defects may appear in the board or to their relative positions.

GRADING BY "ENUMERATED DEFECTS" WRONG PRINCIPLE

Obviously, the cut-up value of the board is vitally affected by the relative positions of the defects, as possibly these may combine so as to render the board next to useless from the standpoint of the waste entailed in cutting to certain requirements. From this a simple and positive statement can be made and, indeed, such a statement crystallizes the thought into specific terms as to the inadequacy of the present grades. Bearing in mind the example just given and illustrated diagrammatically in Fig. 1, the following should be clear: *No grade, be it a top grade, an intermediate grade or the lowest grade, should be determined by the enumerated allowable defects without regard to the location of those defects in the board.*

Considering an intermediate grade, "No. 1 Common" may be briefly reviewed. In this grade "clear face cuttings" are specified, which must be not less than 4 in. wide by 2 ft. long, or 3 in. wide by 3 ft. long, neither being given the preference, thus making it impossible to figure a minimum for the larger-size cutting. There are factory interests that from the nature of their manufacturing should use a relatively low-grade lumber, but in such lumber they must be assured of certain cuttings. If these interests require the 4-in.-wide cutting, this grade of No. 1 Common, just defined, would give them absolutely no assurance of a minimum percentage of 4-in. cuttings. The present grade of No. 1 Common allows the 4-in. cutting, but places no restriction that cuttings shall be wider than 3 in. in order that the board shall make the grade. The grade specifies "clear face," but some utilities require the opposite faces to be clear and others all four faces clear.

For the reason that grading on the present basis does not take cognizance of such factors, that are vitally important from the utilization standpoint, there is the ever-increasing tendency to buy top-grade lumber. This, of course, includes all the clear lumber produced from the tree, so the purchaser of this grade knows that he is getting the best lumber produced and, accordingly, that he stands a better chance of covering his cutting requirements.

The next step in the process is for the consumer to feel the increasing cost of this top-grade material and to come to the conclusion that the cost is too great. He will then cast about for a substitute for hardwoods. Hardwoods have certain inherent qualities that are possessed by no other raw material. These qualities are fully tested and a substitution of other qualities cannot produce the same finished article. Ultimately, therefore, the burden is passed on to the public. By the same process, the producer of hardwoods finds a ready sale for the high-grade lumber from the log, but is burdened with as much as from 40 to 50 per cent of low-grade material. The latter is very difficult to sell and is sometimes impossible to move, so that through exposure over a period of many years at the sawmill it may become next to worthless.

ECONOMIC SEASON FOR HIGH PRICE OF HARDWOODS

In the economic analysis of this condition, the top grades under the present basis must carry the burden of the low grades; otherwise the hardwood producing plant could not function. And it must be borne in mind that the production of low-grade hardwoods entails the same proportionate expense that is incurred in the production of top-grade.

Under the circumstances just cited and because of the lessening of the supply of high-grade hardwoods in our forests, the price-trend of this material must be upward; there can be no alternative. Corrective measures are therefore in order and are urgently needed. The pro-

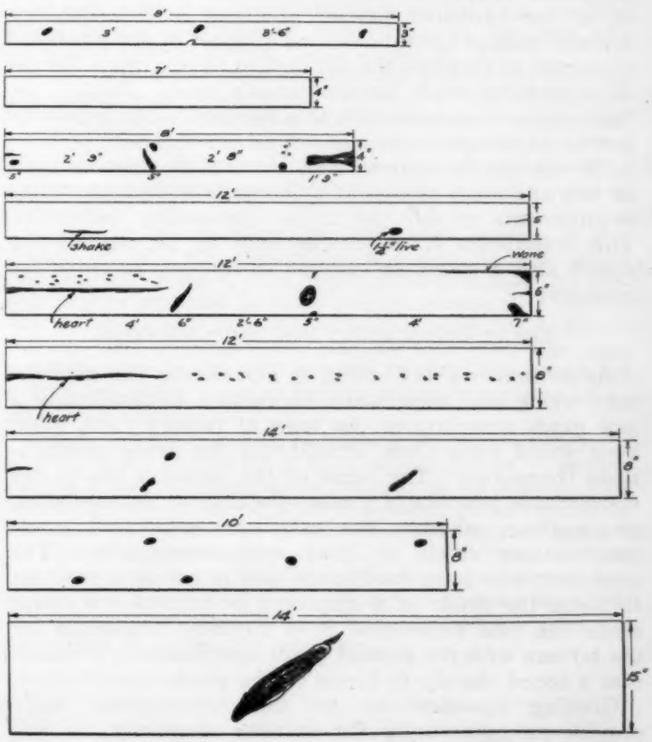

FIG. 2—WIDELY VARYING CLASSES OF BOARDS ALL FALL IN THE NO. 1 COMMON GROUP

ducer of hardwoods senses the situation and seeks relief on the same fundamental ground that the consumer complains of in his inability to realize from a shipment of lumber in the face of oppressive waste.

The drawing reproduced in Fig. 1 shows the boards that are graded as either No. 1 or No. 2 Common and that in Fig. 2 illustrates the wide range in types or classes of board admitted to the one grade of No. 1 Common. A study of these will indicate clearly the economic deficiency of a grade structure that places all these boards under a like classification. Fig. 2 answers very clearly the query as to why today two shipments of No. 1 Common hardwoods on identical orders may give factory yields varying up to 30 per cent.

It should be clear that the deficiency of the present system lies in the failure to classify the board on the yield basis. In the present scheme the entire board is the grading unit. In factory lumber the board is to be cut up; hence it is fundamentally wrong to base the grade upon the entire board as a unit. The factory cut-up size should be the grading unit and the manner in which factory sizes are contained in the board should determine the grade, for on that basis is the value of the board determined for factory purposes. *It is the usable material in the board that determines its value to the factory and, accordingly, this is the only basis upon which a grading structure should be built.*

Corrective Effort to be Applied in Safe and Sane Way

The present grades are not based upon this simple and unassailable fact. However inadequate the present structure may be, it cannot be abandoned until a new vehicle is available and that vehicle has been tested and found to be thoroughly true. It is in the development of this new structure that the Hardwood Manufacturers' Institute, an association restricted in membership to hardwood producers and representing over 35 per cent of all the hardwood lumber produced in this Country, seeks to engage the interest and support of the hardwood consumer to expedite the developing of maximum service in accordance with the consumer's need. Within the limitations of manufacture this represents the maximum service to the producer also.

To achieve the desired end, a sound foundation must be laid and must consist of a thorough knowledge of the requirements of all the major consuming industries. This knowledge is obtainable only by an engineering search and a thorough canvass in cooperation with the consumer.

Great Opportunity for Simplification

Aside from a great saving in eliminating the unnecessary waste and such would be certain in developing a new grade structure on the basis of factory yield, great good would come from simplifying the grade specifications themselves. The curse of the industry lies in the disreputable practice of grade substituting, manipulating or juggling, wherein the existing complicated grade specifications shield or cloak misrepresentation. The consumer who buys hardwoods and is not in a position to judge the grade of a shipment is helpless. Without study and field experience, it is virtually impossible for the layman with the present grade specifications to recognize a board rapidly in terms of the grade classification.

Grading specifications on the factory-yield basis provide an opportunity for extreme simplicity, as will be apparent from the following example, in which the existing grade of No. 1 Common is given, there being printed directly underneath it a simplified grade specification on the yield basis that has been suggested as to form only. This form is even more complicated than it might be in grades where there is need of only one determining factor instead of two. Simple specifications of this character are readily understandable and would preclude disreputable practices in the substitution of hardwood grades. Both the producer and the consumer of hardwood would be provided with a simple yardstick measure of lumber values, and the easy application of such standards would do much to remove many of the difficulties that are experienced in interpreting and in the functioning of the present grades.

No. 1 Commons

(Present Grade-Basis)

Bright sap is no defect in the common grades.

Widths 3 in. and greater, not to exceed 5 per cent of 3-in. widths.

Lengths 4 ft. and greater, but not more than 10 per cent, may be 4, 5, 6 and 7-ft. lengths.

Pieces 4 and 5 ft. long, must be clear.

Pieces 3 and 4 in. wide, 6 and 7 ft. long, must be clear.

Pieces 3 in. wide, 8 to 16 ft. long, must work 66⅔ per cent clear-face in not over two cuttings; no cutting to be less than 3 ft. long by the full width of the piece.

Pieces 4 in. wide, 8 to 16 ft. long, must work 66⅔ per cent clear-face in not over two pieces in cuttings 2 ft. and over long by the full width of the piece.

Pieces 6 ft. long, 5 to 8 in. wide, may have one standard defect; pieces 6 ft. long, 9 in. and over wide, may have two standard defects.

Pieces 5 to 7 in. wide, 7 to 11 ft. long must work 66⅔ per cent clear-face in not over two cuttings; pieces 5 to 7 in. wide, 12 ft. and over long, must work 66⅔ per cent clear-face in not over three cuttings.

Pieces 8 in. and over wide, 7 to 9 ft. long, must work 66⅔ per cent clear-face in not over two cuttings; pieces 8 in. and over wide, 10 to 13 ft. long, must work 66⅔ per cent clear face in not over three cuttings; pieces 8 in. and over wide 14 ft. and over long, must work 66⅔ per cent clear-face in not over four cuttings.

No cutting to be considered which is less than 4 in. wide by 2 ft. long, or 3 in. wide by 3 ft. long.

Simplified Specifications

(Suggested as to Form Only)

Primary Determining Factor.—4 in. and over wide, 48 in. and over long, clear face.

Secondary Determining Factor.—4 in. and over wide, 24 in. and over long, clear-face. Each board must contain not less than 50 per cent primary factor and yield a total of not less than 75 per cent of primary and secondary factors combined.

Central Committee on Lumber Standards

The Hardwood Manufacturers' Institute has created an engineering department to canvass the requirements of consumers and to cooperate with all the elements properly concerned. In the automotive field, work has already been initiated with the Society of Automotive Engineers, specific contact having been made with the Passenger-Car Body Division of the Standards Committee.

Progressing from these intimate field-contacts, the entire lumber standardization program proceeds to the Central Committee on Lumber Standards that has headquarters in the City of Washington. This is an executive body that will act upon approved recommendations when all the elements are in harmony. The ultimate step in this process will be the establishing of American

lumber standards; and this comprehensive structure will include all the lumber specifications for all species, both softwood and hardwood, factory and building woods. The entire fabric will be constructed as a standardized instrument that will be worthy of the term "American Lumber Standards" and will take its place with other American standards, registered in the Bureau of Standards under the Department of Commerce. Hardwoods for factory use, in which, as has been said, the automotive industry stands second in the volume consumed, will play an important part in developing the American lumber standards and there is no question that specifications on the yield basis will rule by the general consent of all concerned.

TRAINING AUTOMOTIVE MECHANICS

AT a conference held by the Federal Board for Vocational Training, in Detroit, on Nov. 29, 1922, the trend of the discussion was influenced mainly by an outline of pertinent subject-matter that had been prepared and circulated in advance.

First consideration was accorded the questions of what type of automotive repairman is in greatest demand; whether assemblers who have learned how to install repair parts or mechanics possessing higher qualifications are the more satisfactory; why any general machine-shop experience may or may not be an asset to a prospective automotive repairman; and which of the following lines of work have value in training automotive mechanics: blacksmithing, sheet-metal work and machinist's bench-work including chipping, filing, scraping, fitting, lining-up and the like.

Two opposing opinions were expressed; one, that in addition to his ability as an assembler the prospective mechanic should have only the bench-work end of the machine-shop work included in his instruction; the other, that a good grounding in general machine-shop practice induces more workmanlike methods of repairing, is conducive to greater pride of the workman in his job and gives him a good basis for future progressive efficiency. A representative of the horological system outlined the three grades of standardized ratings for watch-repairmen as determined under the direction of the Bureau of Standards at the City of Washington. Reference was made to the fact that certification of graduate mechanics should be made as rigid as possible, the present driving licenses being quoted to show how such a system can be abused and robbed of its value. A suggestion was made that vocational schools might be graded according to their qualifications, presumably by the Federal Board, and a brief discussion centered about whether it is planned to educate men to be foremen or to be general mechanics. Further discussion related to the possibility and desirability of having the vocational schools endeavor to specialize their pupils, for part of the course, on certain vehicles to be designated by the students.

While no definite conclusion was reached regarding the subjects outlined in the foregoing paragraph, or even any preponderance of opinion expressed in regard to any one of them, the tendency seemed toward giving the pupil as much instruction and experience in the trades closely allied to the actual repairing of automobiles as might be possible; but that extremes should be avoided, such as foundry work, precision grinding and the like. It is believed that the time available, usually 12 to 18 months, is insufficient, and that the ordinary pupil would not be able to assimilate more than a limited amount of such instruction.

The general opinion appeared to be that an automotive mechanic should have some knowledge of the properties of the materials upon which he works, such as iron, steel, brass, aluminum and some other metals; also, that pupils should be given sufficient information about the heat-treating of steel to enable them to realize the danger of overheating parts when making repairs and that, possibly, in regard to minor parts, the student should be made able to approximate proper methods after he has ascertained from the manufacturer or some other source what heat-treatment, if any, should be employed for the work in hand.

With regard to whether a student should be taught any of the commercial methods of the garage business, a practically unanimous opinion was evidenced to the effect that instruction should be given in garage-business fundamentals and several opinions were given in favor of including also some instruction in salesmanship, so that the student will be assisted in selling his services to an employer and enabled to establish better contacts with customers.

Elementary instruction only was believed necessary in specialized lines such as gas-flame welding, radiator repairing, and tire and storage-battery work, the view being that actual working conditions should be the exclusive consideration in the school shop and that highly specialized tools are apparently losing ground in the larger repair-stations. However, suggestions were made as to means of obtaining shop equipment for the schools from motor-car manufacturers and sales agencies at better rates.

Concerning the extent to which technical courses in the related sciences, such as drawing and mathematics, should be considered an integral part of a course of instruction, what the subject-matter should comprise and upon what it should be based, it was thought desirable to equip a lecture-room in such a school and to have lectures delivered that would tend to tie pure science of this character to actual shop practice, but no definite opinions were expressed as to how deep into the related sciences such a course should go.

It has been stated that an important part of the training of a welder is the development in him of a "job conscience," and the questions were raised whether a similar qualification would be desirable for an automotive mechanic and, if so, how to develop it; but the only suggestion made was that machine-shop training would aid materially in such an accomplishment.

In view of the short time available each evening, it was brought out that any comprehensive course, such as giving shop training on repair work to beginners or making the evening courses available to those who can secure shop experience on the job as a part of their regular work, would extend over too long a period; but, starting with men, who have had little or no mechanical training, 6 weeks was considered entirely too short a period for the adequate training of an automotive mechanic.

Regarding certain phases of training and whether they can be given better in a school or in an outside garage or service-station, it was explained that a majority of the schools bring in outside work in an endeavor to simulate garage operating conditions and that some schools maintain separate garages for this purpose.

It remained an open question as to what agency could best assume the responsibility for setting up training standards and job specifications, if these were deemed advisable for application to automotive repairmen.

Vol. XII January, 1923 No. 1

A Double-Reflector Type of Head-Lamp

By Bruno Heymann[1]

Illustrated with Photographs and Diagrams

THE Committee on Motor-Vehicle Lighting of the Illuminating Engineering Society conducted a number of road tests with head-lamps in 1918 for the purpose of establishing a set of rules as a basis for the approval of road-lighting devices. As a result of the experiences gained, these rules were revised Feb. 9, 1922, and were published.[2] These rules were approved at a recent meeting of the motor-vehicle administrators of 10 States. This action undoubtedly will pave the way for their establishment as the recognized standard in this Country by the American Engineering Standards Committee.

It seems that the efforts of the American designers have been directed mainly to the production of new lenses, tilting mechanisms and other devices applicable to existing ordinary head-lamps representing on the whole makeshifts. Very little attempt has been made to approach the problem from an entirely new angle. It has also been advocated recently that the highways of the Country be illuminated as a solution of the lighting problem. Such an installation would be not only very wasteful, as at least some of the lamps would have to burn all night, traffic or no traffic, but also very undesirable from the standpoint of the driver, since lack of contrast on the road would prevent the quick vision of obstacles. Furthermore, the passing at regular intervals of light-sources of high intensity would produce periodic contraction and dilation of the iris and finally eye-strain. These objections are patent, not to mention the most formidable argument against it, which is the high cost of installation and maintenance that the taxpayer would oppose most strenuously. The only alternative to this proposal is to have each vehicle its own light-source for the illumination of the road.

Glare Problem in Europe

A brief resume of the glare problem in Europe shows that the authorities abroad are struggling with the same difficulties, striving to find a basis for a satisfactory head-lamp specification. Since the end of May, 1922, a new law has been in operation throughout France, limiting the vertical spread of the beam projected by vehicle head-lamps to 1 m. (39⅜ in.) from the level ground. Head-lamps that do not comply with this requirement are prohibited. The operation of powerful automobile head-lamps, undimmed, within the city limits, is also prohibited. For this reason many automobiles, particularly taxis, can be observed driving through the streets of Paris equipped only with old-fashioned oil-lamps. The situation in London is similar. After all, no satisfactory head-lamp has been developed in France to date.

No uniform law with respect to lighting vehicles is in existence in Germany at present. In a general way local ordinances have one provision in common, namely, that in driving through city streets the head-lamps must be switched off entirely or must be dimmed. The head-lamps must also be dimmed on the country highways in meeting other vehicles. This is rarely done, however. Very elaborate constructions for tilting and sliding spherical mirrors and cups around and over the lamp-bulbs are being marketed in an attempt to solve the glare problem. It is safe to say that these mechanisms are rarely operated, as their manipulation is left to the discretion of the driver.

The Departmental Committee on Lights on Vehicles of the Ministry of Transport of Great Britain[3] states that it has conducted very extensive experiments on the best standard electric head-lamps and devices put forward today as a solution of the dazzle problem. The Committee defines glare or dazzle as "such an intensity of light reaching the eye of an observer as to cause material interference with his normal vision, under the conditions existing at the time of observation." Correct as this definition is, it appears to me that it should be somewhat extended, since the degree of dazzle in addition is a function of the total flux of light reaching the eye of an observer as well as a function of the contrast in intensity between the light-source and its background.

These well-known facts have been confirmed again in a series of head-lamp experiments carried out recently by Dr. G. Gehlhoff and Dr. H. Schering at the Goerz plant in Germany. In their report,[4] the authors state that the permissible intensity of a 9-in. head-lamp on a well-lighted street is 27,000 apparent cp.; while, on a dark country highway, an intensity of only 900 apparent cp. constitutes the extreme allowable upper limit.

In the British tests,[5] more than 100 lamps and devices were thoroughly tested in the laboratory, some of them being also subjected to the road test. This test, originated by the Royal Automobile Club and standardized by the British Engineering Standards Association, was approved and adopted by the Ministry of Transport. The result of these tests revealed the fact that not one wholly satisfactory and practicable lamp or dimming device to avoid glare has been discovered. The Committee reaches the conclusion that "prevention of dazzle under every condition and circumstance is at present unattainable with a safe driving-light."

The report furthermore points out that dimming devices cause such a reduction of intensity in the driving light as to render them unsuitable for the faster moving vehicles, and that a much more promising solution of the problem lies in modifications of the design of the head-lamp by which improved control over the beam of light will be attained.

In Appendix 1 of the report, the Committee finally records a detailed specification of head-lamps for the information and guidance of designers and manufacturers, in which the two essential characteristics of head-lamps are

(1) A penetrating beam for distant illumination
(2) A beam of diffused light to illuminate the fore-field

Major A. Garrard, who has been associated with the tests made for the Ministry of Transport Committee[5] in

[1] M.S.A.E.—Mechanical engineer, dean of the California School of Mechanical Arts, San Francisco.

[2] See *Transactions of the Illuminating Engineering Society*, vol. 14, p. 64; see also S.A.E. Handbook, vol. 1, p. B7.

[3] See Third Interim Report, Sept. 30, 1921; see also *Transactions of the Optical Society* (London), vol. 23.

[4] See *Transactions of the Optical Society* (London), vol. 23.

[5] See *Transactions of the Optical Society* (London), vol. 23.

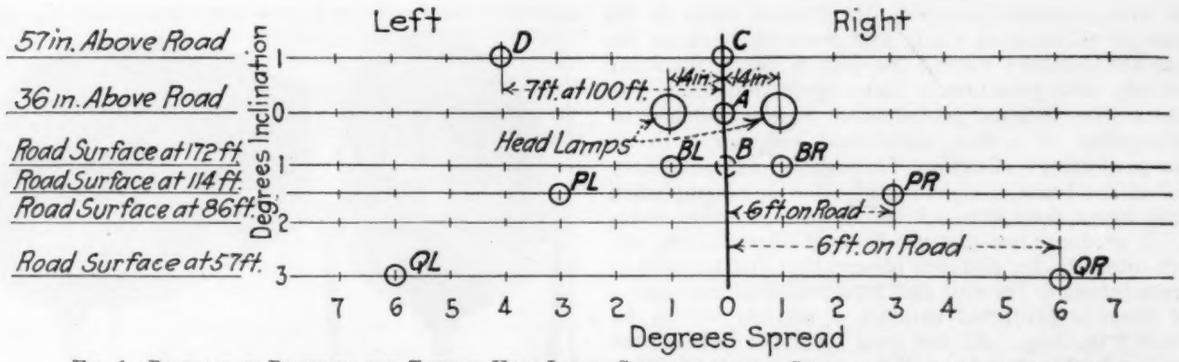

FIG. 1—DIAGRAM OF POSITIONS FOR TESTING HEAD-LAMPS SPECIFIED BY THE SOCIETY OF AUTOMOTIVE ENGINEERS

an excellent paper on Motor-Car Headlights presented to the Illuminating Engineering Society in London,* March 17, 1921, discusses the rules formulated by the American Illuminating Engineering Society, analyses the glare problem and finally states that the essentials of a head-lamp that will keep the light below the eyes and will at the same time give sufficient illumination for safety both of the vehicle and of others on the road are

(1) A fairly wide but relatively dim beam, with a well defined upper limit, 3 or 4 ft., from the ground so as to illuminate hedges, ditches and the road in front of the car

(2) A central broad but shallow penetrating beam of much greater intensity with a well defined upper limit of 3 or 4 ft. from the ground for distance observation

(3) A diffused or subdued beam having a very wide angle, and corresponding to the light obtained directly from the filament

In conclusion, therefore, the question of a satisfactory driving light is not one merely of intensity but also one of distribution.

* See *The Illuminating Engineer* (London), vol. 14, p. 920.

Lenses or systems of lenses, owing to their optical characteristics, cannot efficiently produce the results demanded by Major Garrard, and it seems that head-lamps constructed of mirrors and combinations of mirrors give more promise of success. It is evident that, to attain high efficiency, the total flux from the light-source must be collected and controlled so that a large part, the primary rays, will be projected in a brilliant, shallow beam into the distance, and the remainder, the secondary rays, utilized for fore-field and side-field illumination.

ANTI-GLARE HEAD-LAMP

In thus reviewing the situation, here and abroad, the outstanding fact is revealed that no practical solution of an all-round satisfactory, anti-glare head-lamp has yet been discovered. It is the purpose of this paper to describe a new and most promising device invented and developed by an engineer in San Francisco. This new design represents a radical departure from the beaten path of head-lamp construction and approaches in a marked degree the ideal driving-light outlined in Major Garrard's paper.

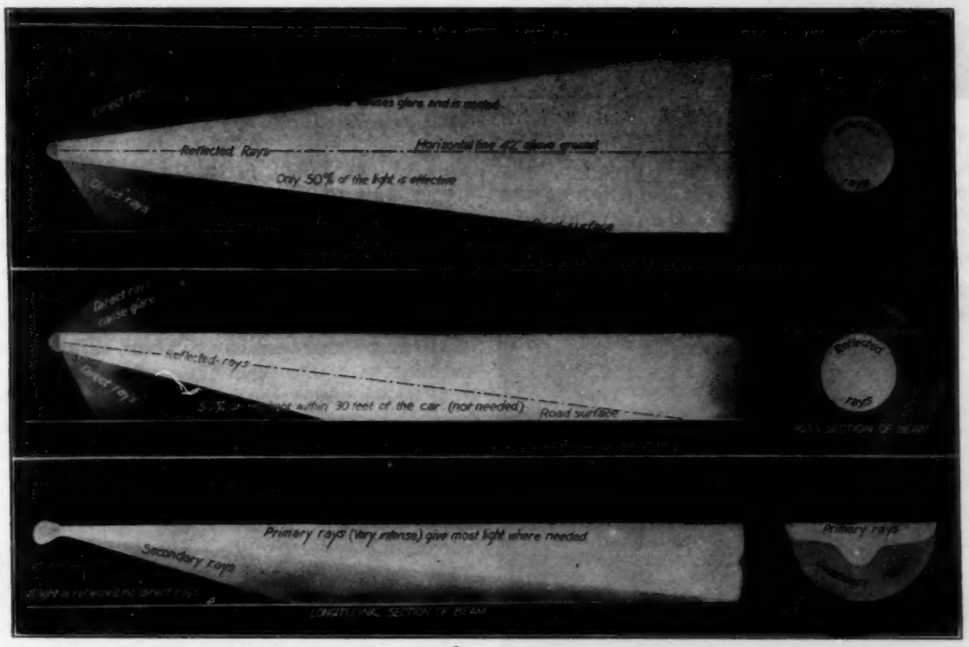

FIG. 2—COMPARISON OF THE CHARACTERISTICS OF A BEAM ISSUING FROM AN ORDINARY FIXED PARABOLOID REFLECTOR UNDER NORMAL CONDITIONS AND WHEN DEFLECTED EITHER MECHANICALLY OR OPTICALLY AND THAT ISSUING FROM A DOUBLE-REFLECTOR HEAD-LAMP

The anti-glare head-lamp in its present form is the outcome of painstaking study and research work on the part of the designer during the past 6 years, involving an entirely new principle in light projection using two reflectors. Its unusual performance is accomplished by a combination of a 6-in. paraboloid reflector with an oblique projecting reflector. Through an extraordinary control of the beam, a light distribution is accomplished that has never been attained before in a practical head-lamp. It produces two distinct fields of illumination, one of high intensity for distance observation and another of moderate intensity for side and fore-field illumination.

The beam is projected through a vertical slit ½ in. wide and 4 in. long. All the rays from the light-source pass through this opening. None is intercepted. They are projected almost entirely below a horizontal plane, 42-in. above the ground.

Fig. 1 is a diagram of specified test positions for testing head-lamps recommended by the Society.[1]

Fig. 2 gives the characteristics of the beam issuing from an ordinary fixed paraboloid reflector at the top of the illustration, the same beam deflected either optically or mechanically in the central portion and that of the double-reflector lamp at the bottom. We are familiar with the objectionable features of the beam coming from the ordinary head-lamp, with its maximum intensity in the axis of the beam, with its glare and its waste of the upper half of the light flux. The tilted reflector, in an attempt to reduce glare, illuminates the roadway with a maximum intensity too close to the car, where it is not needed, and where it is very harmful to the vision of the driver in his endeavor to observe distant objects. The tilted beam illuminates these remote objects only with a reduced intensity where high intensity is paramount.

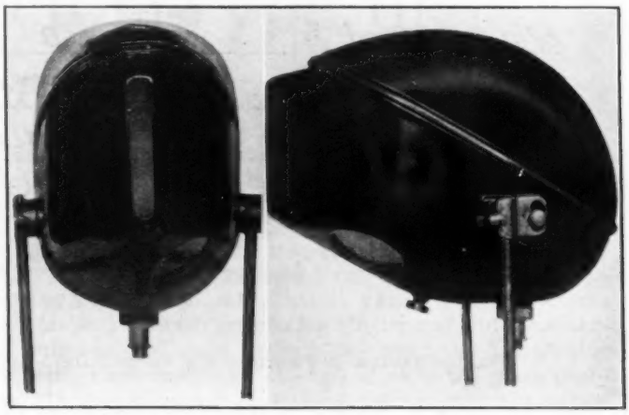

FIG. 5—FRONT AND SIDE VIEWS OF THE DOUBLE-REFLECTOR LAMP

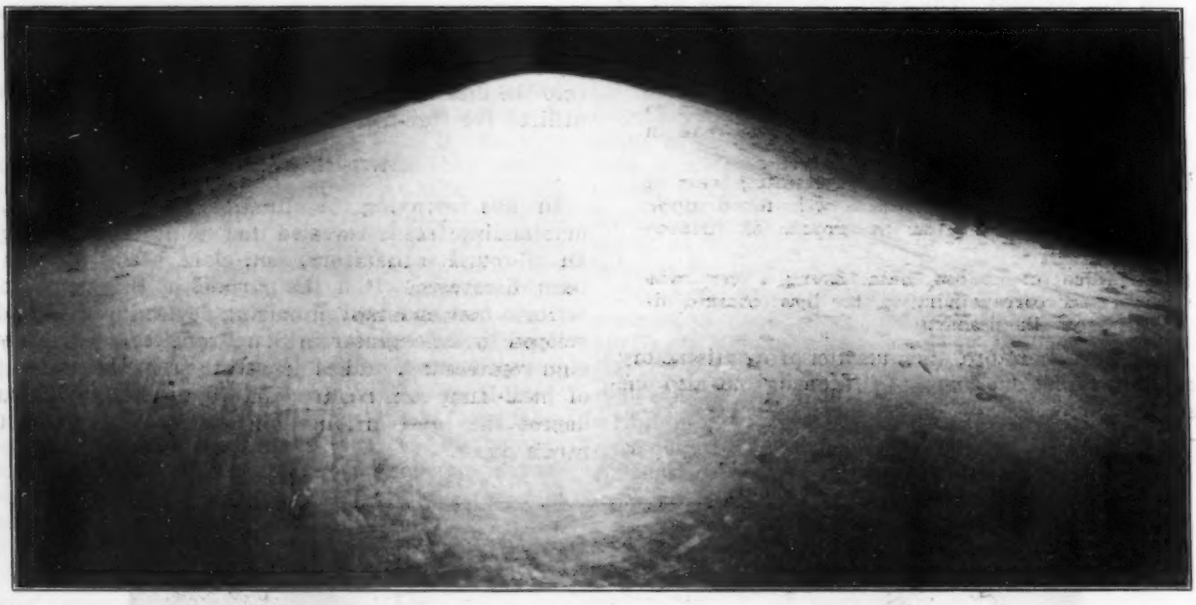

FIG. 3—ILLUMINATION OF THE ROAD WITH A DOUBLE-REFLECTOR HEAD-LAMP

[1] See S.A.E. HANDBOOK, vol. 1, p. B8.

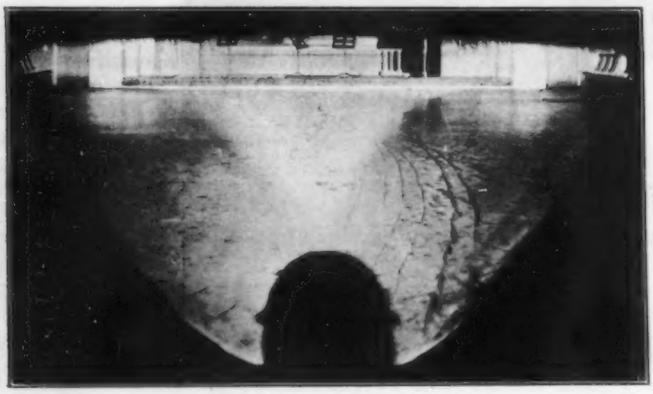

FIG. 4—INTERIOR OF A THEATER ILLUMINATED WITH A DOUBLE-REFLECTOR LAMP

In the beam projected by the double-reflector lamp the maximum intensity is distributed uniformly across the top of a broad field of illumination, as if the wasted upper half of the ordinary beam had been folded back over the lower half, doubling its intensity. The primary rays of high intensity run parallel to the roadway, spreading approximately 50 ft. in a distance of 100 ft. The secondary rays of lower intensity illuminate the regions directly in front and to the side of the car. Additional side and fore-field illumination is obtained by an independent, small side-lamp integral with the housing. This side-lamp can be used also as a parking light.

Fig. 3 shows the illumination of the road. The ob-

server stands directly in front of the head-lamp, looking toward it. The small vertical white speck in the distance indicates the aperture of the lamp. The distance is approximately 100 ft. Note the absence of glare and the distribution of the light on the roadway by a single lamp.

Fig. 4 shows the interior of a theater, looking toward the stage. The position of the observer is directly behind the lamp. Note the 50-ft. spread of the beam on the wall 100 ft. distant, the moderate fore-field illumination and the high intensity of the beam across the horizontal top line, 42 in. above the ground. Note also the uniform height and the sharp definition of beam.

TABLE 1—HEAD-LAMP TEST DATA

		Apparent Candlepower at Test Positions							
		A	*B*	*C*	*D*	*E*[8]	*F*[8]	*P*	*Q*
1922 Limits of the Illuminating Engineering Society and the Society of Automotive Engineers	Max	1,800	7,200	800	.....	1,200	1,200	5,000	2,000
	Min.	6,000	.....	2,400	800	.....	.....	.....	.....
Head-Lamps and Lenses Tested		Results of Tests							
Double-Reflector Head-Lamp[9]		9,000	13,000	800	700	12,500	16,500	19,000	4,000
Lens T[10]		4,200	10,750	2,150	790	1,390	6,750	.....	.....
Lens U[10]		2,050	7,550	910	540	1,525	14,000	.....	.....
Lens V[10]		3,900	11,000	1,900	535	1,015	5,400	.....	.....
Lens W[10]		2,000	6,100	970	510	815	9,750	.....	.....
Lens X[10]		3,350	6,700	1,900	640	1,010	5,750	.....	.....
Tilting Reflector Y[11]		1,165	1,640	990	495	1,120	8,230	.....	.....
Tilting Reflector Z[12]		3,710	4,930	2,110	915	.....	.....	.....	.....

Note:—Many states limit bulb-candlepower for head-lamp lenses in addition to limiting the tests, but such limitations do not apply to the double-reflector head-lamp; any size of bulb can be used, because glare is eliminated.

[8] Test positions *E* and *F* were omitted in the 1922 Revised Rules.

[9] Tests were conducted by the engineering department of the National Lamp Works of the General Electric Co., in the fall of 1920. Each head-lamp was equipped with a 6-in. paraboloid reflector and a 36-cp. bulb. The lamps were 42 in. apart, and readings were taken at a distance of 45 ft.

[10] Data supplied by the division of motor vehicles of one of the states that has adopted the rules of the Illuminating Engineering Society. The lenses are of leading American makes.

[11] Each lamp was equipped with a 9-in. paraboloid reflector and a 14.5-cp. bulb. The lamps were 26 in. apart.

[12] Each lamp was equipped with an 8⅝-in. paraboloid reflector and a 16.4-cp. bulb. The lamps were 25¼ in. apart.

In Table 1, test data are given comparing the lamp with double reflectors and a few of the best known approved lenses and devices with the requirements formulated by the Illuminating Engineering Society and the Society of Automotive Engineers. These rules specify the use of 21-cp. bulbs in making head-lamp tests. All State traffic-laws in approving lenses and other devices also impose limitations upon the candlepower of bulbs to be used. These provisions are made with a view to limiting glare. The double-reflector lamp practically eliminates glare, such limitations become obsolete and bulbs of any candlepower can be used.

The peculiar body shape of the double-reflector lamp is revealed by a study of Fig. 5, which gives front and side views of the newest design. The interior is readily accessible by a removal of the air-tight cap, held in position by one clamp. The front slit, as well as the side opening for the parking light, is plainly visible.

Fig. 6 gives a vertical section through the lamp in which *a* is a 6-in. paraboloid reflector, and *b-c* an inclined reflector, redirecting and reshaping the light beam as it is received from *a*. Both reflectors are manufactured from sheet brass, highly polished and silvered; they are mounted so that they constitute a rigid, complete unit. All light rays meet in a vertical line at the aperture *d* before leaving the housing *e*. The housing is pressed from light-gage sheet-steel stock and black enameled. The focusing adapter *f* permits a minute adjustment of the position of the filament in all directions. A small spherical reflector used for side lighting is indicated at *g*. The cover *k* is held securely in position by the clamp *h*. The weight of one lamp complete is approximately 7 lb.

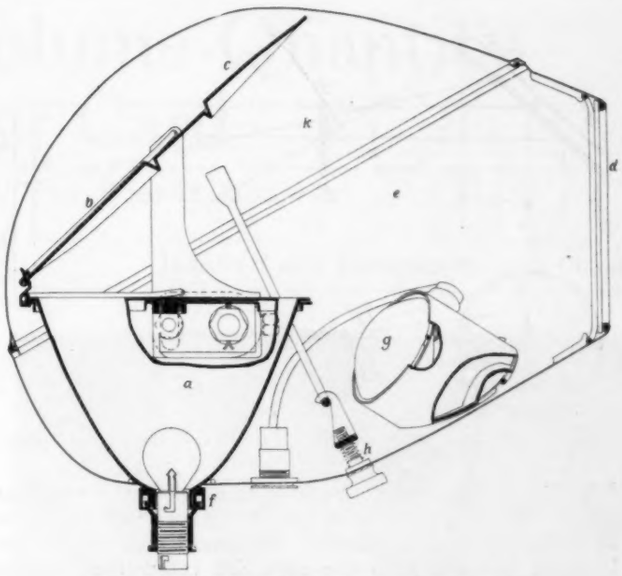

FIG. 6—VERTICAL SECTION THROUGH THE LAMP

FIG. 7—PHANTOM VIEW OF THE LAMP

For the purpose of illustrating the operation of the lamp and its light control, a phantom view is shown in

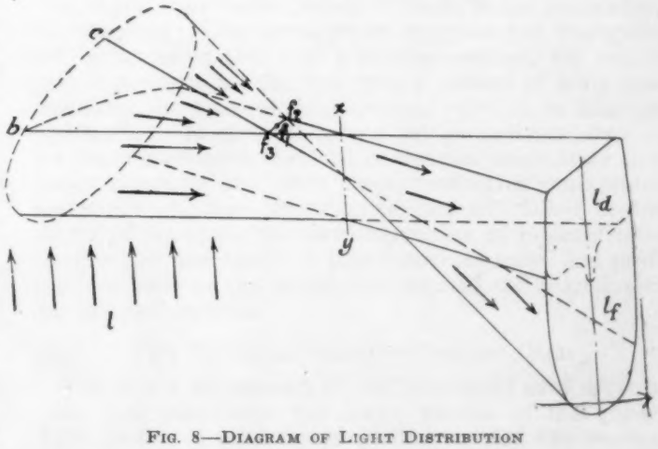

FIG. 8—DIAGRAM OF LIGHT DISTRIBUTION

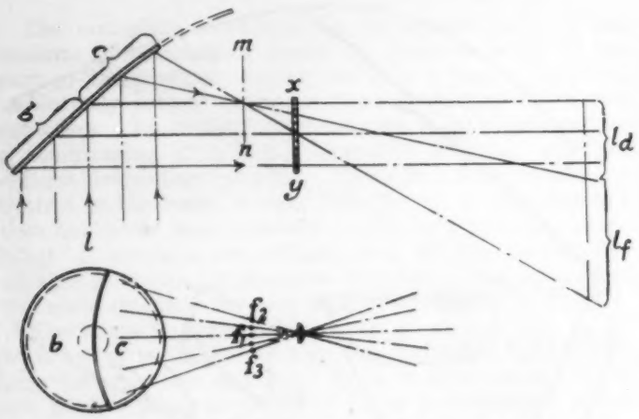

FIG. 9—MERIDIAN SECTION AND PLAN VIEW OF THE HEAD-LAMP

Fig. 7, a pictorial representation in Fig. 8 and a meridian section with a plan view in Fig. 9. Here l represents the flux of light as it is leaving the lower vertical reflector. In their upward path the rays are intercepted by the inclined reflector b-c, redirected and reshaped into two distinct fields of intensity l_d and l_f. This latter reflector is a surface of rotation, whose generatrix consists of two distinct branches and whose axis is x-y. The lower branch b is an oblique straight line forming an angle of approximately 45 deg. with the vertical axis x-y and generating the surface of the frustum of a cone. This reflecting surface collects the bulk of the light rays and, in projecting them, furnishes the powerful, penetrating beam l_d for distance observation as it would appear on a vertical screen placed in front of the lamp. The upper branch c is an arc of a parabola, having its axis at m-n and its focus at f_1. The reflecting surface generated by rotating this parabolic arc about the axis x-y produces the field l_f of moderate intensity for the region directly in front and to the side of the car in addition to strengthening the field l_d. It will be observed that the light rays from c cross first on a horizontal circular arc f_2-f_3 and then again on the vertical line x-y.

The mirror, as manufactured, differs slightly from this theoretically correct surface to compensate for the absence of reflected light from the base of the bulb and for the purpose of radiating direct rays from the source.

The lower portion of the surface b is actually generated, not by a straight line but by an arc of a parabola of a large parameter, whose vertical axis and focus are to be found to the left of the mirror. This modification has not been shown in the illustration for fear of obscuring a perfect understanding of this description.

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The Pressure-Volume-Quantity Indicator-Card

By Thomas Midgley, Jr.[1] and Robert Janeway[2]

Indiana Section Paper *Illustrated with* Photographs and Charts

IN the case of the internal-combustion engine, where virtually every separate portion of explosive mixture behaves differently, the usual thermodynamic interpretations of the pressure-volume indicator-card, as applied to steam engineering, have little value. In internal combustion, the pressure-volume diagram is of value only as an expression for the product of the force exerted upon the piston-top times the distance through which the piston moves.

The paper begins with the fundamental phenomena and develops therefrom a diagram such that each fuel-mixture particle can be properly exposed for analysis during the process of combustion. This is termed the pressure-volume-quantity card, and it is described in detail and illustrated. An extended consideration of its surfaces follows, inclusive of mathematical analysis. Work and losses are discussed, including the mathematical derivation of the so-called pseudo-hysteresis loss, the gross and the net work done; a comparison is made with the pressure-volume card; the indicated thermal efficiency relation is treated; a representation of detonation is presented; and the important points of the paper are summarized.

THE application of thermodynamics to the internal-combustion engine has been notably lacking in beneficial results. This is primarily because of the fact that the so-called thermodynamics of internal combustion has been an extension of the science as it is applied to steam engineering, rather than a development strictly applicable to internal combustion and based upon its fundamental phenomena.

To be more specific, the pressure-volume indicator-card, which is ordinarily considered as a part of thermodynamics, is essentially nothing more than elemental energetics representing in a graphical manner the product of force times distance. In steam engineering, certain curves of this diagram have been found to have thermodynamic interpretations. These interpretations have been sufficiently reliable to yield the necessary information as to the heat interchanges occurring during the process of converting heat energy into mechanical energy. This is due to the fact that any one portion of steam within the steam-engine cylinder can rightfully be considered as behaving in identically the same manner as any other portion. But, in the case of the internal-combustion engine, where virtually every separate portion of explosive mixture behaves differently, such interpretations have no value. Consequently, in internal combustion, the pressure-volume diagram is of value only in its fundamental application; that is, as an expression for the product of the force exerted upon the piston-top times the distance through which the piston moves. So far as internal combustion is concerned, all attempts to convert this card into useful thermodynamics have been unsuccessful.

Fig. 1—Pressure-Volume-Quantity Card for Normal Combustion

The purpose of this paper is to begin with the fundamental phenomena and develop therefrom such a diagram that the behavior of each particle during the process of combustion can be exposed for analysis properly. Unfortunately, there is at present a lack of quantitative fundamental data to draw from, and this very lack of data can be attributed to the fact that heretofore such a system as we present in this paper has not been available. Therefore, to develop the theoretical hypotheses, it has been found necessary to make many quantitative assumptions. It has been found expedient also to use some approximate mathematical expressions where the true expressions are extremely difficult, if not impossible, to integrate. This development is presented therefore, not in the belief that it is a finished product, but rather that it presents for the first time a method of truly representing the thermal relationships existing in internal combustion. It gives at once a better understanding of the thermodynamics involved in internal combustion in a qualitative way and, with these qualitative conceptions before the engineer, quantitative data will follow in due course of time; so, the thermodynamics of internal combustion may eventually be placed upon as sound and profitable a basis as the thermodynamics of steam engineering has proved to be.

The Pressure-Volume-Quantity Card

Fig. 1 is a photograph of the theoretical card development that constitutes the major portion of this paper. This card is a solid figure plotted so that the vertical

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axis represents pressure, P; the left-to-right axis, volume, V; and the front-to-rear axis, progressive portions of the charge, Q.

On such a diagram the work done by an individual particle is represented by the cross-section of the figure as cut by a plane of Q = a constant. The work done by or upon the entire charge is represented by total volumes. This particular diagram is a theoretical card of 100-per cent volumetric efficiency with perfect scavenging. It makes no allowance for the mechanical shortcomings of the engine, valve mechanism and ignition. However, an effort has been made to make due allowance for the heat losses to the jacket occurring in an engine.

At point A of the card in Fig. 1, the piston is at bottom dead-center. It is assumed that the cylinder contains a charge of theoretical mixture proportions at atmospheric pressure and temperature plus a quantity of exhaust gas equal to the clearance volume. Compression takes place along the compression line $A - B$, the piston arriving at top-center B where ignition takes place. The piston is assumed to halt at top-center during the process of combustion. The first particle of gas that burns expands from B to C, this expansion being caused by three-quarters of its heat of combustion. The remaining one-quarter of the heat of combustion of the particle is assumed to be unavailable on account of equilibrium conditions between the uncombined atoms from the fuel and the products formed by the partial combustion of the particle under consideration. The foregoing discussion considers simply the first particle of mixture that has entered combustion. As an aggregate of fuel particles undergo combustion, it is evident that the pressure will be raised; so, by the time one-tenth of the total mixture has been burned, the unburned portion of the mixture will have been compressed adiabatically until its pressure has been raised to the point D. When the next particle of the gas undergoes combustion, it burns along the line $D - E$. This process is continued until combustion of the charge is complete.

While the gas ahead of the flame-front is being compressed adiabatically, the portion of the charge that has been burned is also being raised to the same pressure; but, in the latter case, the character of the compression is not adiabatic, because the gas is losing heat to the cylinder-walls by radiation and conduction. The information on this compression is meager and far from complete. However, the information that is available indicates that the nature of this compression approaches isothermal conditions more nearly than adiabatic. No material departure from these conditions is made if, in the absence of accurate data, it is assumed that this compression is of such a nature that during its compression the burned gas comes to precisely the same conditions of temperature and pressure as that portion of the gas which has just completed combustion. Thus, the burned gas which has been compressed to F is assumed to be at the same temperature and pressure and to have the same specific volume as the gas at E. This process of compression to the rear of the flame-front continues until all the gas has been burned, when its condition is represented by the line G-H.

Expansion now takes place along a surface, which can be determined by actual indicator-cards, to the end of the stroke, I-J, at which point the exhaust-valve opens and conditions return to atmospheric, thus completing the card.

It may be observed that the burned gas which has been compressed along a near isothermal and expanded along a near adiabatic has thereby caused work to be done upon it. This work is represented by the volume L. Applying an electrical term, this volume, which represents a loss of work, can be called the pseudo-hysteresis loss. The positive volume of the card represents the gross work that has been done by the gas. The difference between the gross work and the pseudo-hysteresis loss is then the net work. This difference should equal the work delivered to the piston and equal the area of the conventional pressure-volume card.

Detailed Consideration of Surfaces

By the very nature of the construction of the diagram, the front and rear surfaces are simple flat planes of equations $Q = 0$ and $Q = 1$, respectively. Assumptions, research and approximations are unnecessary in these cases.

Up to the point at which ignition occurs, the equation governing the compression surface can be determined from an actual indicator-card, and the surface can be extrapolated from this point to the completion of combustion. The general equation for this surface should be $PV^y = K$. No assumptions or approximations are necessary here; but, to justify the extrapolation thoroughly, further research work should be carried on to determine definitely that the unburned gas ahead of the flame-front does compress adiabatically.

The combustion surface is one of the most interesting developments of this theoretical work. It has been shown, in a paper[3] entitled Molecular Movements during Combustion in Closed Systems, presented at a meeting of the Pennsylvania Section in 1921, that the general equation for an intercept of this surface by any vertical plane, Q = a constant, should be

$$P_1 - P = W^2/g\ (V - V_1) \qquad (1)$$

where

g = 32.2
P = the pressure
P_1 = a constant
V = the specific volume
V_1 = a constant
W = the quantity of gas entering the flame-front per unit time

This intercept is a straight line of slope proportional to the square of the reaction velocity. In normal combustion, the departure of this line from the horizontal is so slight that it need not be taken into consideration. It is only when detonation occurs that the slope of this line plays an important part in the diagram. The projection of the combustion surface upon a vertical plane, V = a constant, is determined in the following manner:

Since at any point during combustion the sum of the volumes of the burned and the unburned portions of the mixture must equal the clearance volume, an expression can be derived on this basis relating the pressure to the quantity of fuel burned. The specific volume of the unburned portion of the gas at any pressure is defined by the point on the compression line corresponding to that pressure; likewise, the specific volume of the burned portion of the gas at the same pressure is defined by the corresponding point on the line $G - C$. Then, if Q represents the portion of fuel burned at any point, the clearance volume, C_v, is

$$(Q \times V_2) + (1 - Q)\ V_1$$

where

$V_1 = (K/P)^{\frac{1}{y}}$, the specific volume of the unburnt gas (adiabatic compression)

[3] See The Journal, May, 1922, p. 361.

$V_2 = (K/P)^{\frac{1}{y}} + \Delta V$, the specific volume of the burnt gas

$y = C_p/C_v$, the ratio of the specific-heats of the gas

$\Delta V = C/P$, as derived in equation (3) for constant-pressure burning

Substituting,

$$V_2 = (K/P)^{\frac{1}{y}} + C/P$$

Then,

$$Q\ [(K/P)^{\frac{1}{y}} + C/P] + (1 - Q)\ (K/P)^{\frac{1}{y}} = C_1$$

$$Q\ (C/P) + (K/P)^{\frac{1}{y}} = C_1$$

$$Q = P/C[C_1 - (K/P)^{\frac{1}{y}}] \qquad (2)$$

The actual variation of P with Q, as determined by equation (2), departs so little from a straight-line law that such a relation can be assumed with inconsiderable error. The equation resulting from this assumption is

$$P = mQ + b$$

Where

b = the compression pressure
m = the total pressure-rise during combustion
P = the pressure in pounds per square inch
Q = the fraction of mixture burned

The situation with respect to the combustion surface can be summarized by saying that very little more research work need be done to make a proper determination of this surface. However, the true equation of this surface is a very complex function, because it is a combination of equations (1) and (2); but, during normal combustion and within the limits of reasonable compression-ratios, the equation of this surface departs so little from the simple expression $P = mQ + b$ that the variations can be considered negligible, and the mathematics can be simplified greatly by adopting this expression as the equation of the combustion surface. While only an approximation, its deviation from the true value is less than 0.5 per cent.

Burned-Gas Compression-Surface

In deriving the expression for the burned-gas compression-surface, the first essential is the determination of the percentage of the heating value of the fuel that is available before equilibrium conditions are set up between the uncombined atoms and molecules of the fuel and the products of combustion. This equilibrium condition has commonly been referred to by engineers as dissociation, and a considerable amount of experimentation and numerous calculations have been made by a number of investigators in attempting to settle this much disputed point. There is as yet no agreement of opinion on this subject, and further experimental work is necessary before it can be established definitely. The net result of the theoretical investigation reported in this paper indicates that the mean average of available heat is about 75 per cent of the total heating value of the mixture. The use of this value will yield net results that agree with practice. In constructing this theoretical diagram, therefore, we have assumed that each particle of fuel, upon combustion, liberates 75 per cent of its total heating value.

The next point to determine is the nature of the compression of the burned gases. Careful analysis of the work of Woodbury, Lewis and Canby[4] yields the suggestion that gases to the rear of the flame-front are compressed approximately in accordance with isothermal conditions, giving up heat by conduction to the walls as well as by radiation. This point deserves further investigation. It is important to establish the magnitude of the pseudo-hysteresis loss, because it can be reduced materially by proper engine design and operation. Inasmuch as there are no exact data covering the nature of this compression, we have felt at liberty to select a type of compression that approximates an isothermal and lends itself to simple mathematical analysis. This assumption is simply that, when each particle of fuel has reached the end of its combustion, all the gas previously burned has been so compressed that it has assumed the same conditions of pressure, temperature and specific volume as that of the particle under consideration. In other words, the assumption is that the burned gas is compressed along the same curve as that which limits the expansion of the burning particles, and which can be seen from its equation, $V = C/P + (K/P)^{\frac{1}{y}}$, to be nearly an isothermal.

The derivation of the equation based upon these assumptions is as follows: If H equals the heat added during constant-pressure burning,

$$H = WC_p\Delta T$$

where

C_p = the constant-pressure specific-heat
ΔT = the temperature rise
W = the weight of the gas burned

Then, for any particle, ΔW, and where h equals the available heat per unit weight of the gas,

$$\Delta W \times h = \Delta W C_p \Delta T$$
$$\Delta T = [(V_2/V_1) - 1]\ T_1$$
$$T_1 = (PV_1 \times 144)/R$$

where

P = the pressure in pounds per square inch
$R = 778\ (C_p - C_v)$, the quantity within the parenthesis being the difference between the constant-pressure and constant-volume specific-heats
T_1 = the initial absolute temperature
V_1 = the initial specific volume
V_2 = the final specific volume

then

$$h = C_p(V_2 - V_1)\,[P(144/R)]$$
$$= w \times H\,[C_c - (C_1[p_a/p_s])]$$

where

C_c = the volume of the cylinder
C_1 = the clearance volume
H = the heat value per pound of fuel
h = the available heat per pound of mixture
p_a = atmospheric pressure
p_s = the suction pressure

$$P\ (V_2 - V_1) = hR/C_p \times 144$$
$$= C \qquad (3)$$

But $V_1 = (K/P)^{\frac{1}{y}}$, from the adiabatic compression of the unburnt gas. Therefore, equation (2) for the curve limiting the expansion, and for the compression of the burned gas becomes

$$V_2 = C/P + (K/P)^{\frac{1}{y}} \qquad (4)$$

The expansion surface can be determined from an actual indicator-card, since its intercept with a plane, Q = a constant, corresponds to the expansion line of the indicator-card. In the theoretical card under discussion, the equation of the expansion surface has been assumed to be $PV^y = A$. No further assumptions are necessary and no approximations are used; so, no research work is necessary on this surface.

Work and Losses

The pressure-volume-quantity card will be observed to consist of two separate volumes; the smaller is called the

[4] See The Journal, March, 1921, p. 209.

pseudo-hysteresis loss, the larger being termed the gross work. The difference between these two volumes represents net work, and it should correspond to the area for work on the conventional pressure-volume card. To determine these volumes it is simply necessary to integrate between the bounding surfaces. The pseudo-hysteresis loss will be considered first.

DERIVATION OF PSEUDO-HYSTERESIS LOSS

Basic Equations

$$V = C/P + (K/P)^{\frac{1}{\gamma}} \quad \text{(Compression of the burnt gas. } C = P\Delta V\text{)} \tag{5}$$

$$V = (A/P)^{\frac{1}{\gamma}} \quad \text{(Adiabatic expansion)} \tag{6}$$

$$Q\,(C/P) + (K/P)^{\frac{1}{\gamma}} + (1-Q)\,(K/P)^{\frac{1}{\gamma}} = C_1 \quad \text{(Clearance volume)} \tag{7}$$

For all practical purposes, the pressure varies with the amount of fuel burnt, Q, according to a straight-line law.

$$P = mQ + b \quad \text{(approximately)} \tag{8}$$

where

b = the compression pressure
m = the total pressure-rise during combustion
Q = the fraction of the charge burnt

$$PV^{\gamma} = K \quad \text{(Compression of the unburnt gas)} \tag{9}$$

The loss is determined by the integration of the expression

$$\int_{Q_1}^{Q_2}\int_{P_1}^{P_2}\int_{V_1}^{V_2} dQ\,dP\,dV$$

The limits of the integration are

P = Final pressure
$P_1 = mQ + b$
$Q_1 = 0$
$Q_2 = 1$
$V_1 = (A/P)^{\frac{1}{\gamma}}$
$V_2 = C/P + (K/P)^{\frac{1}{\gamma}}$

Substituting the limits given, we obtain

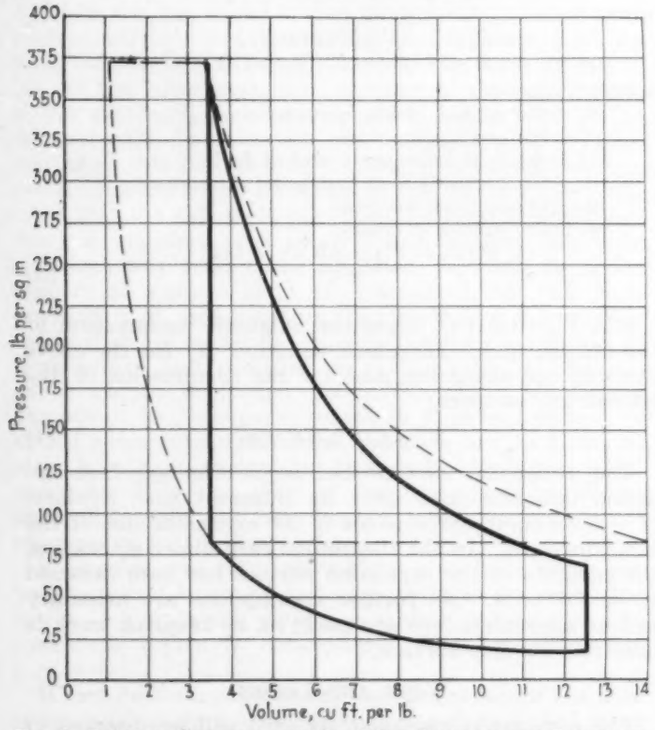

FIG. 2—DIAGRAM SHOWING THE NET WORK DONE BY THE GASES AND THE WORK DELIVERED TO THE PISTON IN A SPECIFIC CASE

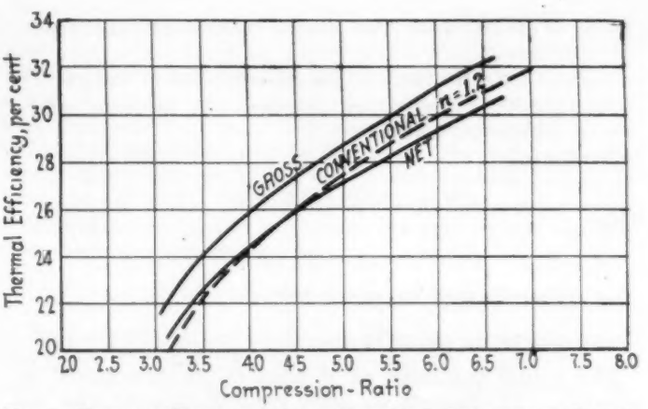

FIG. 3—THERMAL EFFICIENCIES FOR DIFFERENT COMPRESSION-RATIOS AS CALCULATED BY THE EQUATIONS BASED ON THE PRESSURE-VOLUME-QUANTITY CARD

$$\text{Loss} = \int_0^1 \int_{mQ+b}^{P_2} \int_{(A/P)^{\frac{1}{\gamma}}}^{C/P + (K/P)^{\frac{1}{\gamma}}} dQ\,dP\,dV$$

Integrating gives

$$\text{Loss} = (K^{\frac{1}{\gamma}} - A^{\frac{1}{\gamma}})/(-1/\gamma + 1)\,[P_2^{-\frac{1}{\gamma}+1} - (P_2^{-\frac{1}{\gamma}+2} - P_c^{-\frac{1}{\gamma}+2})/(P_2 - P_c)\,(-1/\gamma + 2)] + C - [(CP_c/[P_2 - P_c])\,(\log P_2/P_c)]$$

GROSS WORK DONE

Basic Equations

$$PV^{\gamma} = K \quad \text{(Compression of unburnt gas)} \tag{10}$$

$$PV^{\gamma} = A \quad \text{(Expansion line)} \tag{11}$$

$$P = mQ - b \quad \text{(Combustion surface)} \tag{12}$$

The card can be divided into two parts at the plane P equals compression pressure, and the two volumes thus obtained can be integrated separately. Then,

$$\text{Gross Work} = \int_{Q_1}^{Q_2}\int_{P_1}^{P_2}\int_{V_1}^{V_2} dQ\,dP\,dV + \left[\int_{V_c}^{V_d} (P_c - P)\,dV\right]$$

where

$P = K/V^{\gamma}$
P_c = compression pressure $= b$
V_c = clearance volume
V_d = cylinder volume

The limits used in the integration are

$P_1 = b$
$P_2 = mQ + b$
$Q_1 = 0$
$Q_2 = 1$
$V_1 = (K/P)^{\frac{1}{\gamma}}$
$V_2 = (A/P)^{\frac{1}{\gamma}}$

Substituting the limits, we obtain

$$\text{Gross Work} = \int_0^1 \int_b^{mQ+b} \int_{(K/P)^{\frac{1}{\gamma}}}^{(A/P)^{\frac{1}{\gamma}}} dQ\,dP\,dV + \int_{V_c}^{V_d} P_c dV - (K/V^{\gamma})\,dV$$

Integrating and simplifying gives

$$\text{Gross Work} = (A^{\frac{1}{\gamma}} - K^{\frac{1}{\gamma}})/(-1/\gamma + 1)\,[(P_2^{-\frac{1}{\gamma}+2} - P_c^{-\frac{1}{\gamma}+2})/(-1/\gamma + 2)\,(P_2 - P_c)] + (K - A)/(\gamma - 1)\,D^{\gamma-1}$$

where

D = cylinder volume
P_c = compression pressure
P_2 = final pressure

Since the net work equals the gross work less the pseudo-hysteresis loss,

$$\text{Net Work} = [(A^{\frac{1}{y}} - K^{\frac{1}{y}})\, y/(y-1)]\, P_2^{-\frac{1}{y}+1} + (K-A)/(y-1)\, D^{y-1} - C + [(CP_c/P_2 - P_c)(\log e\{P_2/P_c\})]$$

Pressure-Volume and Pressure-Volume-Quantity-Card Comparison

In Fig. 2, the solid lines represent a conventional pressure-volume card, and the dotted lines represent the

Fig. 4—A Pressure-Volume-Quantity Card Showing Detonation

projection on the plane, $Q = 0$, of the corresponding pressure-volume-quantity card.

The main usefulness of the pressure-volume card, as stated previously, is as a graphical representation of the pressure applied to the top of the piston against the distance through which the piston moves, the area of which is a measure of the work delivered to the piston. On the other hand, the pressure-volume-quantity card represents the work done *by* the gases. It is evident, therefore, that the work represented by the volume of the pressure-volume-quantity card should equal the work represented by the area of the pressure-volume card. Actual calculations upon the special case diagrammatically represented in Fig. 2 give the following result per pound of mixture: Net work done by the gases as obtained from the pressure-volume-quantity card equals 146,200 ft-lb. The work delivered to the piston as represented by the area of the pressure-volume card equals 146,600 ft-lb.

Fig. 5—Pressure-Time Card from an Engine with Normal Combustion

The slight discrepancy between the above two numerical values is representative of the approximations used in deriving the formula for net work on the pressure-volume-quantity card. It is felt that the agreement between these two numerical values is sufficiently close to justify for the present the assumptions and approximations made in this work.

The diagram in Fig. 3 shows thermal efficiencies for different compression-ratios calculated by the equations based upon the pressure-volume-quantity card. The gross as well as the net work is represented. The difference between these two curves represents the pseudo-hysteresis loss. No increase of heat loss due to higher compressions was used in these calculations; so, the increase in the pseudo-hysteresis loss indicates the increasing importance of correct thermal design in engines of higher compression-ratios. The customary efficiency curve, derived by conventional thermodynamics using $n = 1.2$, is shown also in Fig. 3.

Representation of Detonation

Although the primary purpose of this paper is not a discussion of the phenomenon of detonation, the paper would be incomplete without showing how detonation can be represented on the pressure-volume-quantity card. The pressure-volume card does not lend itself to depicting detonation because, at the time detonation occurs, the piston is approximately at top dead-center; the result is that the pressure rise in both normal combustion and detonation is represented as a single straight vertical line at one end of the card. The pressure-time card is a distinct improvement upon the pressure-volume card for studying detonation insofar as it shows a distinct difference between pressure changes in normal combustion and the rapid rise of pressure that occurs in detonating combustion. However, the pressure-time card does not convey a clear conception as to what detonation is. A pressure-volume-quantity card representing detonating combustion is shown in Fig. 4.

Referring to the formula $P_1 - P = (W^2/g)\,(V - V_1)$, it has been pointed out in this paper that the normal reaction velocity, W, is so small that the Q intercept of the combustion surface does not depart materially from a horizontal line. When detonation occurs, this is no longer true; W becomes so large that the departure of a Q intercept of the combustion surface from the horizontal must be taken into account. In fact, this intercept becomes nearly vertical instead of horizontal, as shown in the diagram in Fig. 4. This representation of detonation on the pressure-volume-quantity card is far superior to the representation on the pressure-time card. However, the pressure-time card must be relied upon for tak-

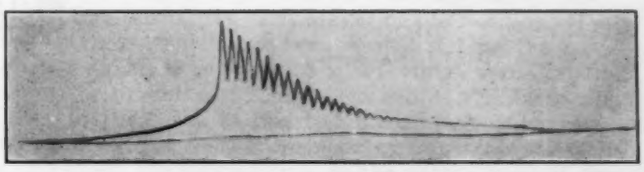

Fig. 6—Pressure-Time Card from an Engine That Was Detonating

ing data on detonation, for which purpose it is far superior to the pressure-volume card.

It is obvious that detonation, as represented on the pressure-volume-quantity card, is a localized phenomenon that only a portion of the fuel charge undergoes. Furthermore, it is obvious from the pressure-volume-quantity card that, during detonation, more mechanical work is obtained from the heat energy of the fuel than in the case of normal combustion, and yet we do not find engines that deliver more work when detonating. The explanation of this is very simple. The increased transformation of heat energy depicted on a pressure-volume-quantity card during detonation is delivered as kinetic energy. The localized pressure is a high-pressure wave that depends for its very existence upon accelerating mass-particles to its rear, and it is the kinetic energy of these mass-particles that is represented by the increased volume shown in Fig. 4. Unfortunately, our engines are not designed to take advantage of these high velocities of the mass-particles. If it were possible to have an auxiliary turbine within the cylinder-head, this kinetic energy could be put to useful work; but, as it is, this energy dissipates itself as a violent surging of the gases, and eventually becomes heat energy.

In substantiation of this explanation, Figs. 5 and 6 are included. These show two photographic pressure-time cards taken from the same engine within 2 or 3 min. of each other. In Fig. 5, the engine was operating on normal combustion. The only surging apparent is of the type normal to such combustion. Fig. 6 shows a card from the same engine when it is detonating. The violent surging shown is a clear indication of the large amount of kinetic energy liberated during detonation that could not be utilized by the piston.

Summary

The important points of this paper are summarized as follows:

(1) A diagram has been presented, which although inaccurate on account of the lack of quantitative data, is nevertheless qualitatively representative of the exact phenomenon of combustion as it occurs in the internal-combustion engine

(2) By means of this diagram, the fallacy of the conventional pressure-volume card as a basis for thermodynamic interpretation of internal combustion is shown

(3) The mechanism of an important loss that has heretofore been improperly understood, the so-called pseudo-hysteresis loss, is revealed

(4) It is shown that detonation can be represented truly upon the pressure-volume-quantity card

THE DISCUSSION

W. G. Wall:—According to the theory advanced in the paper, the expansion of certain particles produces a higher pressure and, when the knock is eliminated, a certain amount of power that would otherwise be available is lost.

Thomas Midgley, Jr.:—No. To have a high-pressure wave, there must be a high concentration of particles in the wave-front. Consequently, the remainder of the charge will have its pressure reduced slightly. However, that is not where the loss of power during knocking comes from. The loss of power is a secondary effect. During detonation, the flame-front is radiating more heat than otherwise, and it is because of this extra loss of heat to the water-jacket that we fail to get as much power out of an engine that is knocking as we do from this same engine when the knock is eliminated by the addition of an anti-knock material.

R. W. A. Brewer:—Have you satisfied yourself as to what causes the noise of detonation?

Mr. Midgley:—Dr. H. C. Dickinson explained that, at least to my entire satisfaction, when he showed that the high-pressure high-velocity wave striking the side of the cylinder-head would cause the cylinder-head to vibrate as though it had been struck with a hammer and would, consequently, give off the typical noise that we hear during detonation.

Mr. Brewer:—The thickness of the cylinder would have an effect upon the character of the sound of the knock, would it not?

Mr. Midgley:—Yes.

P. J. Dasey:—Have you made any investigation as to what effect different spark-plug positions have on detonation?

Mr. Midgley:—The spark-plug position has some effect. I cannot lay down a definite set of rules for the best location of spark-plugs. To my mind, the spark-plug position in the Knight engines seems to be ideal. This is, however, physically impossible to obtain in a valve-in-head engine.

G. T. Briggs:—Have you any definite idea as to the practical limit to which one can go in raising compressions?

Mr. Midgley:—I feel confident that a 7 to 1 compression-ratio will be very close to the practical limit in increasing compressions. We found that we gained so little in going above a ratio of 7 to 1 that it was not worth the trouble.

A Member:—How do you explain the increased tendency of kerosene to detonate, compared with gasoline?

Mr. Midgley:—We simply know that it is true. In the equation $W = KD^nT^m$, the factor K must be determined experimentally. In the paraffin series, we have satisfied ourselves that K increases with a decrease in volatility. In the case of the aromatics, K diminishes with a decrease in the volatility. How it behaves in other series of hydrocarbons, we do not know at the present time, but we are working on the problem.

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Internal-Combustion Engine Characteristics Under High Compression

By J. H. Holloway,[1] H. A. Huebotter[2] and G. A. Young[3]

Annual Meeting Paper *Illustrated with* Photographs and Charts

THIS paper is a report of a series of tests conducted during the summer of 1922 by the authors at the Engineering Experiment Station of Purdue University. The work consisted of research into the operation of internal-combustion engines under comparatively high compression on ordinary gasoline without detonation. The compression-ratio of the engine was 6.75 and the compression pressure was 122 lb. per sq. in., gage. The ingoing charge was passed through a hot-spot vaporizer and thence through a cooler between the carbureter and the valves. Jacket-water temperatures between 150 and 170 deg. fahr, were carried at the outlet port of the jacket.

The theory held by the authors as to the causes of detonation of the combustible charge is presented briefly. The source of the two phases of detonation encountered in this work is believed to be overheated areas in the combustion-chamber. The methods of combating these hot spots are given in detail, and the special equipment applied to the engine to accomplish the desired result is described. The effects of load, speed, compression-ratio and mixture-ratio are studied, and curves showing the variation in the engine characteristics due to each factor are submitted. All tests were run without a trace of detonation.

The results of the work point to the following conclusions:

(1) Automobile engines designed to give a uniform cooling of the combustion-chamber walls will permit the use of much higher compression-ratios than those employed at present, with a consequent gain in engine power and economy

(2) The fuel-air ratio for maximum economy borders upon the lean limit for reliable combustion

(3) For maximum power, the mixture-ratio need not exceed 0.075 lb. of gasoline per lb. of dry air

(4) With concentrated heating of the mixture at the carbureter outlet, a local temperature of 125 deg. fahr. is sufficient to give good operation at full throttle. Cooling to 100 deg. fahr. at the valves is desirable if the compression pressure of the engine is high. At part loads, a hot-spot temperature of 175 deg. fahr., with or without intermediate cooling, will give good results with lean mixtures

THE factor that controls the thermodynamic efficiency of an internal-combustion engine is the expansion-ratio. In the standard automotive engine, this is practically equal to the compression-ratio, which determines the pressure of the charge at the time of ignition. To secure the benefits of greater expansion, therefore, the detonation that results when compression pressures are high must be eliminated.

The compression-ratio that can be used with any given fuel is limited by the tendency of the engine to detonate under certain conditions of operation. If the gain in efficiency, due to the increased expansion of the hot gases, is great enough to offset the additional expense and care required in the design, construction and maintenance of the engine to obviate detonation, it is evident that the higher compression-ratio is desirable. It is the purpose of this paper to set forth the means adopted by us to overcome detonation under high compression-pressures, to show the gain in power and economy and to present such incidental characteristics of engine performance as apply to engine and accessory design.

Detonation, as the term is applied ordinarily in internal-combustion-engine operation, is accompanied by one of two characteristic kinds of knock. The sharp metallic "ping" that is most commonly encountered in automotive engines appears to originate from too early ignition of the compressed charge. This knock, if due to excessive spark-advance, is eliminated by proper timing of the ignition. If retarding the spark produces no diminution of the knock, the preignition is probably due to some overheated spot within the combustion-chamber. We know of no means of distinguishing between these two sources of preignition except by manipulation of the spark-timing.

A second kind of detonation occurs after ignition has originated from the electric spark and is characterized by a dull, heavy thud. Such a knock is evident at times in both high and low-compression engines. It is apparently an intermediate stage between auto-ignition and normal combustion. With high compression-pressures, ignition that is timed to occur after the dead-center may be followed by this heavy thud when the engine is thoroughly heated-up. Automobile engines with the usual compression-ratio show the same trait, when the spark is retarded after a period of preignition, due to an early spark or to excessive carbon deposit. One form of detonation may merge into the other or may disappear entirely, depending upon the condition of normal engine operation. Sufficient spark-lag will eliminate the heavy pounding, but at the expense of both power and economy.

The following theory seems to offer a reasonable explanation of this second phenomenon. The charge is ignited by the spark and burns at a normal rate of flame propagation during the early stages of the reaction. The unburned gas, which has been heated during the compression stroke, is compressed still further by the expansion of the products of combustion behind the flame-cap. Unless the heat due to this latter compression is transferred to the combustion-chamber walls as rapidly as it is generated, the temperature of the entire mass of the gas rises to the ignition point and explodes instantaneously, producing a hammer-blow effect in both shock

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[2] M.S.A.E.—Research assistant, Purdue University, West Lafayette, Ind.

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and sound. In support of this hypothesis is the experimental evidence that the knock ceases when certain suspected high-temperature areas are subjected to additional cooling. The absorption of heat from the combustible charge is thereby facilitated and auto-ignition is prevented.

The foregoing description and explanation of the two types of detonation point to the following sources as being most open to suspicion. Preignition due to auto-ignition before the passage of the spark may originate from an excessive temperature of either the spark-plug core or electrodes, the exhaust-valve, the center of the piston-head or from any other point in the combustion-chamber that is not cooled sufficiently. Detonation that follows normal ignition may be due to any of the above sources except the first. Hence, careful attention to the cooling of the entire combustion-chamber is essential to the suppression of detonation.

In addition to the elimination of hot-spot areas, it is advisable to maintain a low entrance-temperature of the fuel-air mixture. The temperature of the compressed charge must never exceed the ignition point of the fuel before the passage of the spark and, to prevent the charge from being overheated by compression in the cylinder, the initial charge-temperature must be reduced to the minimum consistent with uniform distribution to secure the thermal advantages of a high compression-ratio.

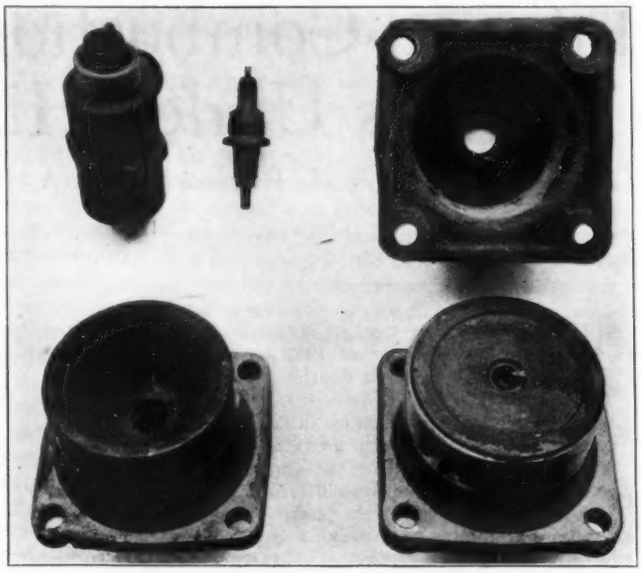


FIG. 2—SPECIAL CYLINDER-HEAD EQUIPMENT USED IN CONNECTION WITH THE TESTS

HIGH-COMPRESSION ENGINE PERFORMANCE TESTS

With this conception of the benefits and the detriments inherent in a high compression-ratio, a series of tests was planned to study engine performance under high compression. The objects of the tests were: (*a*) to determine roughly the maximum compression-pressure an automobile engine can carry on standard gasoline under a representative range of operating conditions without detonation, and (*b*) to study the characteristics of the engine when so modified. The difficulties encountered in securing satisfactory operation, the methods used in overcoming them and the reasons for the adoption of these methods are presented herewith.

A compression pressure of 150 lb. per sq. in. was employed at first to accentuate the detonation and to facilitate the location of the sources. Two engines of different types were used in the work. Beginning with engines that were standard in all respects with the exception of the cylinder-head, the load was gradually increased until detonation became apparent. The cylinders in which the knock occurred were studied with reference to the probable factors inducing detonation. These factors were eliminated gradually until the engine operated satisfactorily under all normal conditions encountered on the dynamometer.

Experiments on a six-cylinder poppet-valve engine showed that spark-plugs with porcelain cores and small electrodes were the first sources of preignition. Mica cores with heavy electrodes and water-jacketed shells were required to maintain a practical working temperature of the plug and, in redesigning the second engine,

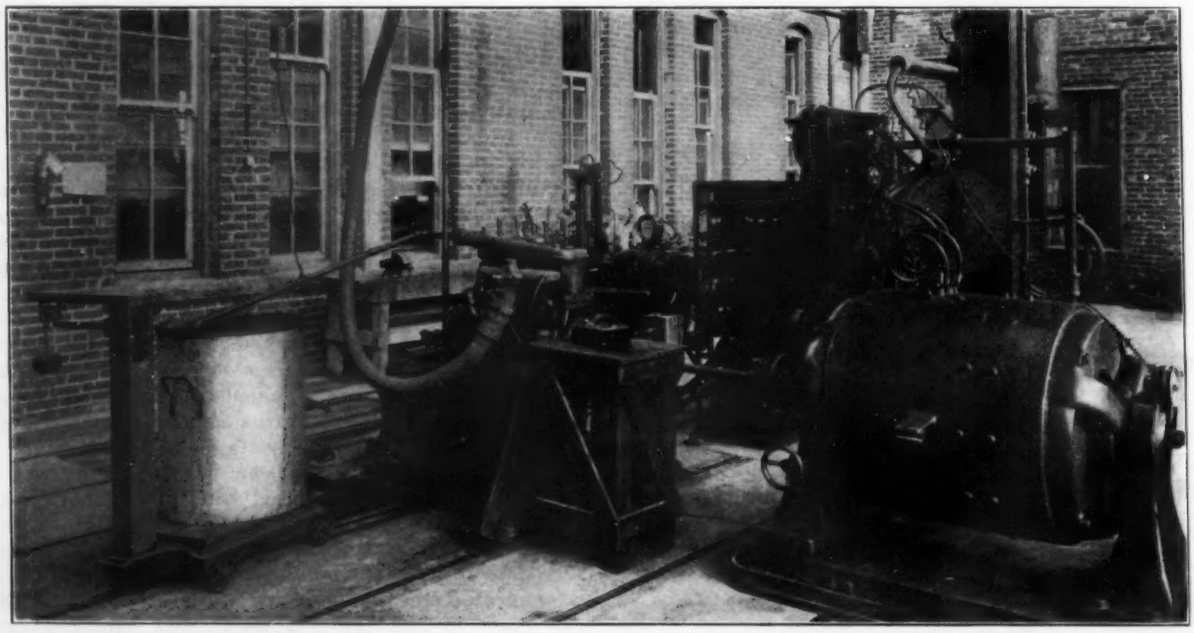


FIG. 1—VIEW OF THE LABORATORY SHOWING THE ENGINE ON WHICH THE TESTS WERE CONDUCTED

this type of plug was incorporated in its cylinder-heads.

After the spark-plug trouble was overcome in the poppet-valve engine, the exhaust-valves showed unmistakable signs of overheating. In the two cylinders that detonated first, the exhaust-valve seats were not completely water-jacketed and could be cooled sufficiently to eliminate preignition only with a water-outlet temperature below 100 deg. fahr. This engine was not a good example of modern poppet-valve design. Trouble from overheated valves was eliminated by changing to a sleeve-valve type of engine on which all further investigation was conducted.

On this four-cylinder engine cylinders Nos. 1 and 4 detonated severely, while Nos. 2 and 3 operated normally. Three possible causes for this effect were considered: (*a*)

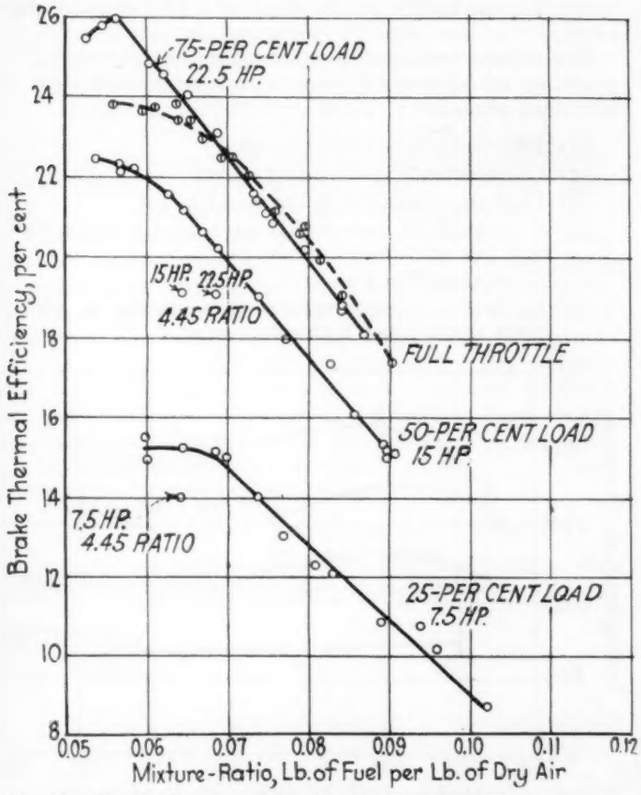


FIG. 3—CURVES SHOWING THE BRAKE THERMAL EFFICIENCY FOR VARIOUS MIXTURE-RATIOS THROUGHOUT THE FULL RANGE OF LOAD

restricted or sluggish circulation of the cooling water at the front and the rear of the water-jacket; (*b*) unequal distribution of the fuel in the charge; and (*c*) arrested scavenging of the center cylinders.

The water-jacket was designed for thermo-syphon circulation with the inlet port between the two center cylinders. In the tests the water was forced through the jacket under pressure from the city supply, which change may have disturbed the uniformity of distribution obtainable with the original system. The single inlet-pipe was replaced by a water manifold with an individual lead to each cylinder to insure more nearly equal cooling. The improvement resulting from this alteration was slight, but the manifold was retained as a safety measure.

The carbureter was at first bolted directly to the cylinder block at the entrance to the water-jacketed inlet-manifold. In this arrangement, good vaporization and distribution were impossible with the gasoline in current use, and a hot-spot vaporizer was therefore placed at the carbureter outlet. The high temperature of the mixture

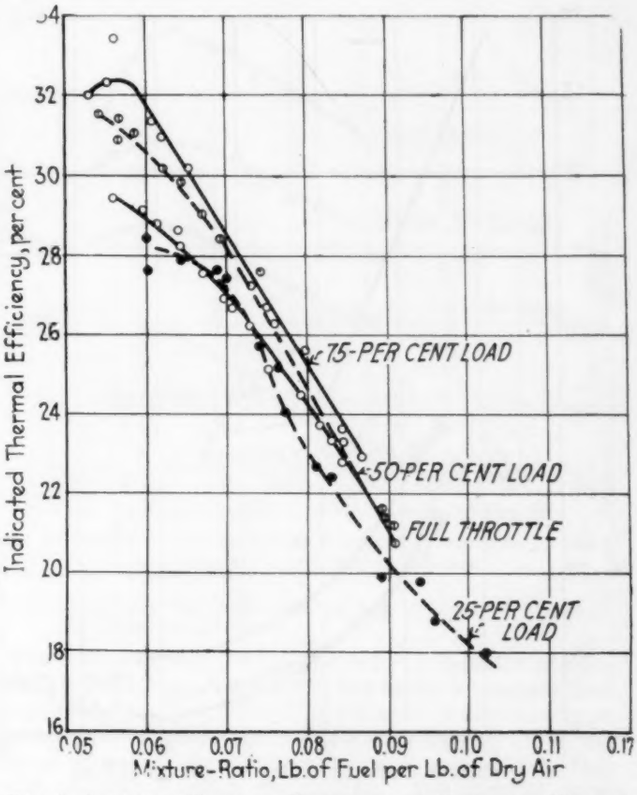

75-PER CENT LOAD
50-PER CENT LOAD
FULL THROTTLE
25-PER CENT LOAD
Indicated Thermal Efficiency, per cent
Mixture-Ratio, Lb. of Fuel per Lb. of Dry Air

FIG. 4—INDICATED THERMAL EFFICIENCY AT DIFFERENT MIXTURE-RATIOS THROUGHOUT THE FULL RANGE OF LOAD

required to secure vaporization of the fuel promoted detonation, and consequently an intercooler was used to lower the temperature of the charge at the valves. The carbureter and the hot-spot were dropped about 24 in. below the inlet-manifold, and a vertical water-jacketed pipe was inserted as a means of cooling the mixture of air and vaporized fuel without causing condensation. The device produced a drop in temperature of about 30 deg. fahr. from an initial temperature of 125 deg. fahr. that previously had been found necessary for good distribution with this vaporizer. The change in the carburetion system permitted the use of higher temperatures at the carbureter outlet and in that way simplified the distribution problem. It was not possible to detect any difference in starting with this lowering of the carbureter position. Cylinders Nos. 1 and 4 continued to detonate before Nos. 2 and 3, although not to such a marked degree. The matter of more thorough scavenging of the center cylinders was not followed-up.

In an engine with suitable spark-plugs and effective

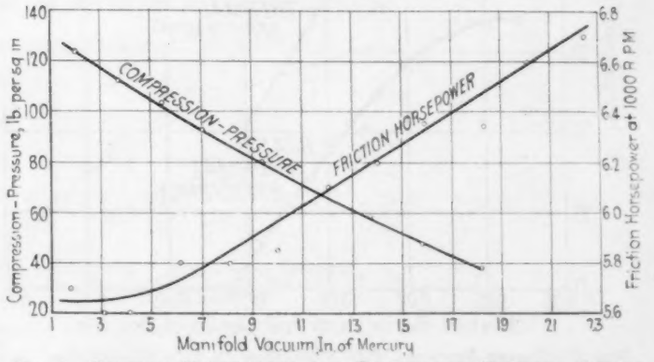


FIG. 5—FRICTION HORSEPOWER AND COMPRESSION-PRESSURE CURVES

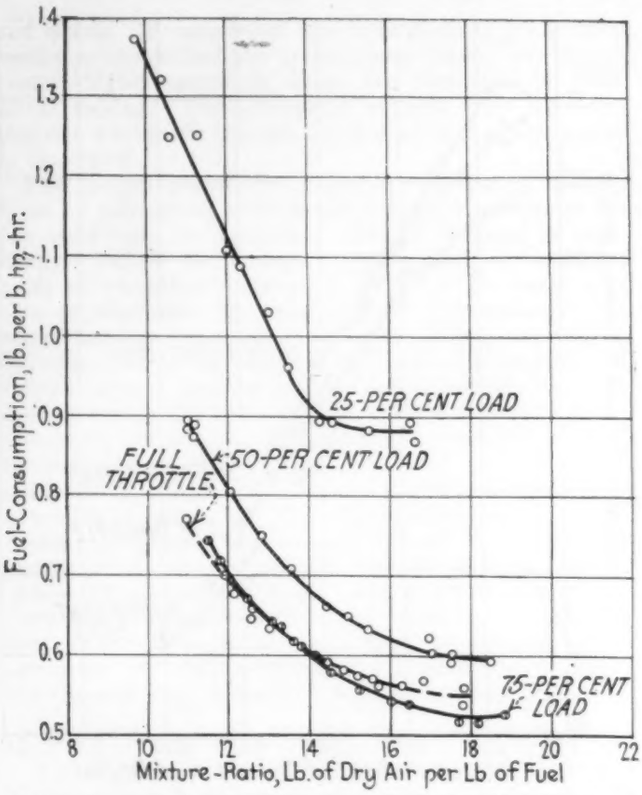

Fuel-Consumption, lb. per b.hp.-hr.
Mixture-Ratio, Lb. of Dry Air per Lb of Fuel
25-PER CENT LOAD
50-PER CENT LOAD
FULL THROTTLE
75-PER CENT LOAD

FIG. 6—SPECIFIC FUEL-CONSUMPTION AT DIFFERENT MIXTURE-RATIOS THROUGHOUT THE FULL RANGE OF LOAD

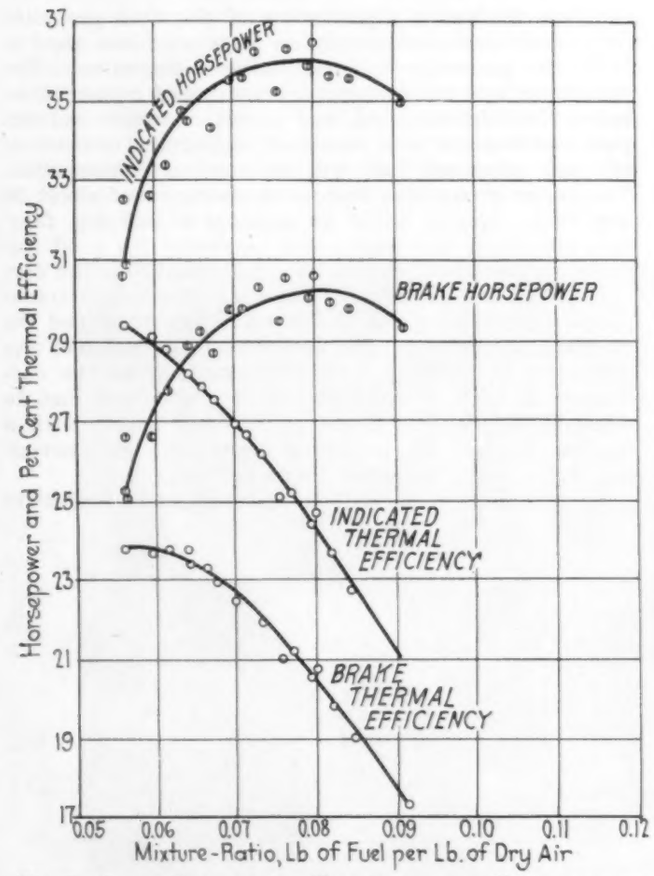

Horsepower and Per Cent Thermal Efficiency
Mixture-Ratio, Lb of Fuel per Lb. of Dry Air
INDICATED HORSEPOWER
BRAKE HORSEPOWER
INDICATED THERMAL EFFICIENCY
BRAKE THERMAL EFFICIENCY

FIG. 7—CURVES SHOWING THE VARIATION IN THE HORSEPOWER AND THERMAL EFFICIENCY AT FULL THROTTLE

cooling of the water-jacketed walls of the combustion-chamber, the temperature of the piston-head is the deciding factor in limiting the compression pressure. This surface, being most distant from the origin of combustion, is in the best possible position to detonate the final portion of the charge ahead of the flame-front. On account of their higher conductivity, aluminum pistons remain cooler than gray-iron pistons, and hence reduce the tendency of the engine to detonate. The original gray-iron pistons were therefore replaced by aluminum pistons that were of substantially the same design.

The breaker of the battery system, with which the engine was originally equipped, failed under the high primary voltage found necessary to assure reliable ignition with high compression at full throttle. This difficulty was overcome by the substitution of a high-tension magneto.

The salient features in the engine that permitted the carrying of high compression under all conditions of operation were

(1) Effectively cooled spark-plugs
(2) Comparatively cool exhaust-valves
(3) Uniform circulation in the water-jacket
(4) A carburetion system that gave good distribution with low mixture-temperatures
(5) Aluminum-alloy pistons
(6) An ignition system capable of producing an adequate spark under high compression

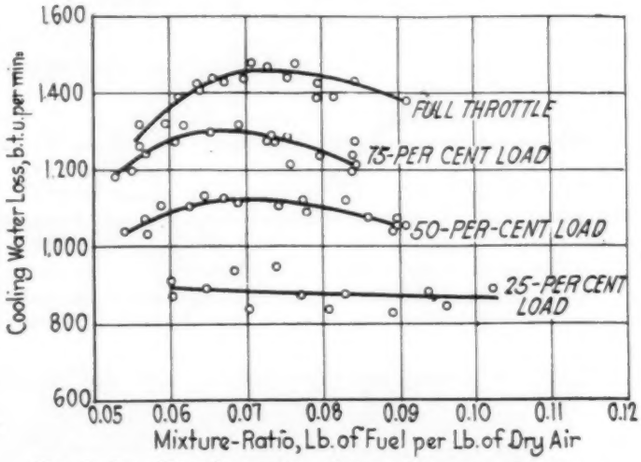

Cooling Water Loss, b.t.u. per min.
Mixture-Ratio, Lb. of Fuel per Lb. of Dry Air
FULL THROTTLE
75-PER CENT LOAD
50-PER-CENT LOAD
25-PER CENT LOAD

FIG. 8—THE HEAT-LOSS TO THE WATER-JACKET AT DIFFERENT MIXTURE-RATIOS

EQUIPMENT AND APPARATUS

The second engine used for this high-compression investigation was a Model 88-4 Willys-Knight of 4⅛-in. bore and 4½-in. stroke, shown in Fig. 1, and slightly modified to adapt it to the abnormal conditions under which it was to operate. As a result of the preliminary work tending toward the suppression of detonation, a compression pressure of 122 lb. per sq. in. was chosen as being the highest practicable for ordinary gasoline. The special heads used in the test to secure this pressure had a compression-ratio of 6.75, as compared with 4.45 in the standard heads and are illustrated in Fig. 2.

A set of standard Willys-Knight aluminum-alloy pistons was substituted for the gray-iron pistons before the tests recorded herewith were carried out. This permitted a greater spark-advance than was possible with the original pistons. Special spark-plugs with shells in direct contact with the cooling water formed the central portion of the high-compression heads and acted as anchorages

for the standard demountable water-jacket cover. A Model SS-4 Splitdorf magneto was installed in place of the battery system. The spark-advance was measured on a graduated sector at the magneto-control lever.

A Model LB-2 Stromberg carbureter was used in connection with a hot-spot vaporizer and a long vertical external inlet-pipe. The vaporizer was heated in such a way as to permit control of the mixture temperature at the entrance to the manifold independently of the engine. After being heated to the point beyond which previous tests had shown no gain in economy was secured, the mixture was cooled in the vertical extension of the manifold. The benefit of this lowered temperature was partly lost by the reheating in the main inlet-manifold, which was surrounded and heated by the jacket water of the cylinder block.

A Sprague electric dynamometer was used to measure the brake and the friction horsepower in the usual manner. The fuel consumed per test was weighed on a beam balance sensitive to 0.01 oz. The air was measured in an Emco No. 4 meter. The inlet-manifold vacuum was determined by a mercury-column gage connected to the manifold between cylinders Nos. 1 and 2. Exhaust temperatures were measured by thermocouples at the No. 4 exhaust-port and at the end of the manifold. Thermometers were inserted at the inlet and the outlet of the water-jacket, at the outlet of the fuel vaporizer, at the top of the manifold water-jacket and at the No. 4 inlet-port. The humidity of the atmosphere was measured by a wet-and-dry-bulb hygrometer. The cooling water for each test was caught in a tank and weighed on a platform scale. Compression pressures were determined with a compressometer consisting of a pressure gage in combination with a chamber and check-valve that screwed into the spark-plug shell.

Test Procedure

Investigations were made to establish the effects of the following four factors in engine characteristics:

(1) Load
(2) Mixture-ratio
(3) Speed

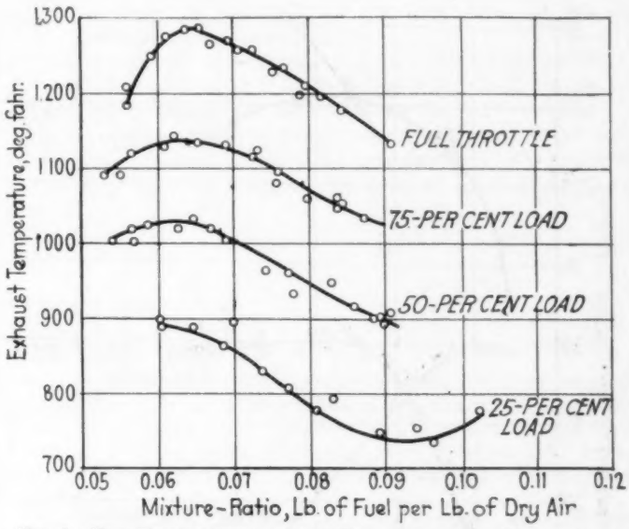


Fig. 9—The Temperature of the Exhaust at the End of the Manifold at Various Mixture-Ratios

(4) Gain in power and economy due to the increase in the compression-ratio

The first two factors were studied at a constant engine-speed of 100 r.p.m. Tests were run under a constant load at approximately 25, 50 and 75 per cent of the maximum power, with various mixture-ratios ranging from the leanest at which the engine would fire regularly to that which showed sluggish combustion. The throttle opening was changed as required to maintain a constant load and speed. Similar tests were run at full throttle and a constant speed with a change in the load as the mixture-ratio changed.

Two series of runs were made with a fixed carbureter-adjustment and at speeds varying from 500 to 1300 r.p.m. In one run, a constant dynamometer-load of 15 lb. was maintained; in the other, the throttle was fully opened.

In all of these tests the spark was adjusted for the maximum power, and the mixture temperature at the hot-spot was held at the point that gave a uniform distribu-

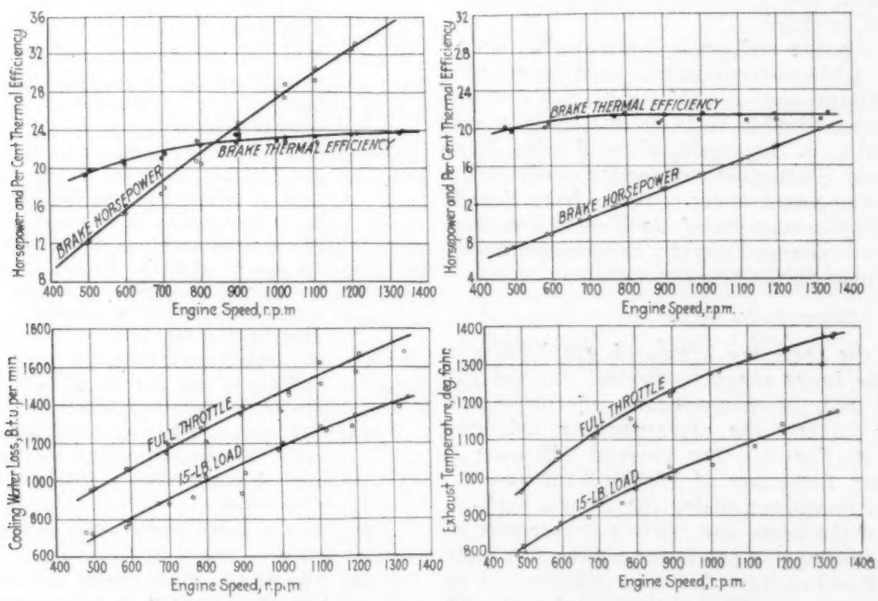


Fig. 10—Curves of Test Results at Different Speeds

Brake Horsepower and Thermal Efficiency at Full Throttle
Heat-Loss to Water-Jacket

Brake Horsepower and Thermal Efficiency at 15-Per Cent Brake Load
Temperature of Total Exhaust

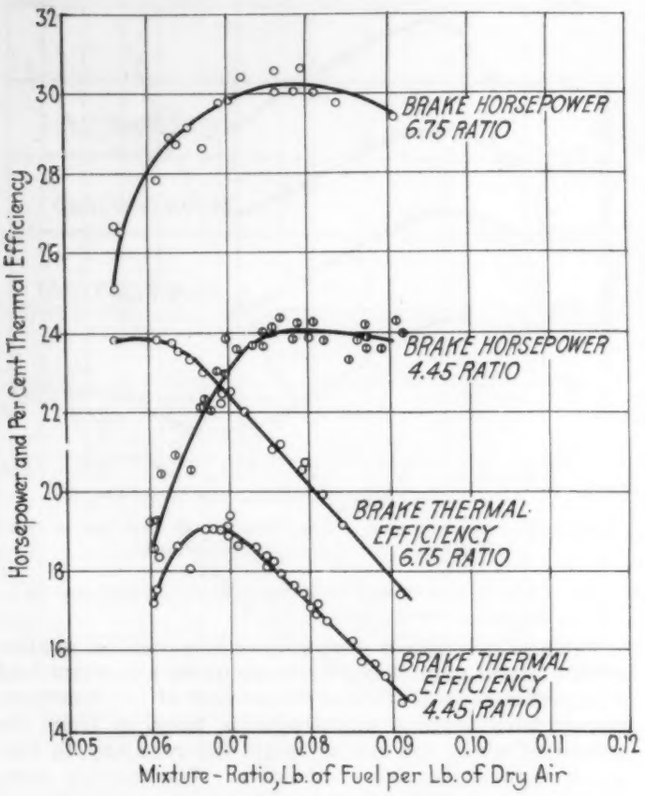


FIG. 11—FULL-LOAD PERFORMANCE WITH HIGH AND LOW COMPRESSIONS

tion without excessive loss in volumetric efficiency. Between the hot-spot and the valves, the maximum cooling of the mixture that was consistent with good operation of the engine was sought. The temperature of the jacket outlet-water was held between 150 and 170 deg. fahr. as nearly as possible, with little adjustment of the flow rate except in the case of the speed series. Fuel, air, cooling water and exhaust temperatures were taken at the points previously described; and load, revolutions per test and time per test were recorded.

The comparative power and brake thermal-efficiency of the 6.75 over the 4.45-compression-ratio are based upon the full-throttle mixture-run at 1000 r.p.m., and data derived from somewhat similar tests on the same engine by O. C. Berry and C. S. Kegerreis, of the Purdue Engineering Experiment Station, in connection with carburetion research. The power curve is taken from Fig. 28 of a bulletin on the Carburetion of Gasoline.[4] The thermal-efficiency curve is changed slightly to correspond with the fuel value assumed in the current tests.

RESULTS OF TESTS

The results of the tests are presented graphically in Figs. 3 to 11. The brake thermal-efficiency for the full range of the load and the mixture-ratio at 1000 r.p.m. is shown in Fig. 3, and the corresponding indicated thermal-efficiency in Fig. 4. The thermal efficiency is based upon a lower fuel-value of 18,900 B.t.u. per lb., and the indicated horsepower under the several loads is taken as the sum of the brake and friction horsepower of Fig. 5. The fuel-consumption per brake horsepower-hour is plotted in Fig. 6 and conveys the same information as Fig. 3 in different terms. The variation of both the power and the thermal efficiency with a change in the mixture-ratio at full throttle is illustrated in Fig. 7. Incidental data showing the heat-loss to the water-jacket and the temperature of the exhaust at the end of the manifold with different mixture-ratios appear in Figs. 8 and 9 respectively.

The influence of the speed upon the brake horsepower, the brake thermal-efficiency, the radiation loss and the exhaust temperature at full load and at approximately half load, are covered in Fig. 10. The carbureter adjustment was not changed during the speed tests and the variation in the mixture-ratio, 0.062 to 0.071, was merely that inherent in the carbureter design under changes in the speed and the load.

A comparison of the power and the thermal efficiency at full throttle with compression-ratios of 4.45 and 6.75 is made in Fig. 11. In Fig. 3 three points taken from the bulletin on the Carburetion of Gasoline[4] are spotted, which indicate the improvement in economy through the full range of engine power.

In the load tests at 1000 r.p.m., the spark-advance was 5 deg. at full load, 12 deg. at 75-per cent load, and 20 deg. at 50 and 25-per cent load. During the full-load speed-tests, the spark-advance varied from 0 to 10 deg. between the speeds of 500 and 1300 r.p.m. At one-half load, the spark lead ranged from 10 to 20 deg. for the same speed variation. No detonation was audible during any of these tests.

CONCLUSIONS

From these tests, which are represented as covering only a narrow range in the field of gasoline-engine operation, the following conclusions are drawn:

(1) Under laboratory conditions, a compression pressure of 120 lb. per sq. in. gage is perfectly feasible when the engine is designed with a full regard for the elimination of the factors that induce detonation

(2) Under service conditions, the same attention to these factors will permit the use of much higher pressures than those common at present

(3) An increase in the compression-ratio results in a marked improvement in the thermal efficiency and the general performance of the engine at all loads and in the maximum power at all speeds

(4) The sources of detonation are the spark-plug, the exhaust-valve, the piston-head and any other portions of the combustion-chamber walls that become overheated

(5) The maximum economy is obtained with the leanest fuel-air mixture giving reliable combustion. Such a mixture varies from 0.060 lb. of fuel per lb. of dry air at 25-per cent load, to 0.057 lb. at 75-per cent load; or, from 16.6 to 17.5 lb. of dry air per lb. of fuel. Hence, for this range of load, the leanest possible mixture is desirable, provided good vaporization is secured. At a full throttle-opening, the maximum power is usually sought at the expense of economy. An 0.075 mixture, or 13.3 lb. of air per lb. of fuel, is rich enough to insure the greatest output

(6) The water-jacket heat-loss reaches its maximum value at approximately the mixture giving the highest power with a fixed throttle-opening, or a 0.075 ratio

(7) The exhaust temperature is highest with a 0.065 ratio

(8) The power of an engine can be increased 25 per cent by a change in the compression ratio from 4.45 to 6.75, provided detonation is absent

(9) With the same change in the compression-ratio,

[4] See Purdue University Engineering Experiment Station Bulletin No. 5.

the thermal efficiency is raised 7 to 12 per cent through a load range of 25 to 100 per cent of the engine power

(10) At full load, a mixture temperature of 125 deg. fahr. at the hot-spot is high enough to give good distribution in a four-cylinder engine with a properly designed hot-spot. If the mixture is cooled below this temperature between the hot-spot and the valves, the performance of the engine is improved. A low limit of 100 deg. fahr. at the valves can be allowed safely.

(11) At partial throttle-opening, a hot-spot temperature of 175 deg. fahr. is not excessive and assists in the use of very lean mixtures for high economy

This investigation is being continued, and it is expected that the final results will be published in a bulletin of the Engineering Experiment Station of Purdue University.

IMPACT TESTING

THE impact test has been used as an acceptance test in certain European specifications. It is not generally adopted in this Country. A committee of the American Society for Testing Materials is working on some form of test suitable for the United States requirements.

The Fremont test requires a test-specimen 30 mm. (1.1811 in.) long with a cross-section of 8 x 10 mm. (0.3150 x 0.3937 in.). It is notched on the broad side 1 mm. (0.0394 in.) deep and 1 mm. (0.0394 in.) wide. The test-piece is laid on supports 21 mm. (0.8268 in.) apart with the notch on the under side and struck a blow with a falling tup midway between the supports. The difference between the energy before and after the fracture gives the amount of energy necessary for fracturing the test-specimen. The weight of the tup, which falls from a height of 4 m. (13.1233 ft.) is 10 or 15 kg. (22.0462 or 33.0693 lb.). A machine of this kind requires considerable height above the floor level, but possesses the advantage of having a high speed of impact.

The Charpy test requires a test-specimen 55 mm. (2.1654 in.) long with a cross-section of 10 x 10 mm. (0.3937 in.). It is notched in the center by drilling a 1.3-mm. (0.0502-in.) hole so that the bottom of the hole is 5 mm. (0.1969 in.) from the edge. The notch is completed by running a 1-mm. (0.0394-in.) slot into the hole. This test-piece is supported horizontally between 40-mm. (1.5748-in.) abutments and is struck by a swinging pendulum at the middle of its length on the side opposite the notch. The energy absorbed in making the fracture is determined from the residual energy after fracture. The weight of the tup is 32.5 kg. (71.6503 lb.). It will be noted that the machining of a notch of this nature requires at least two operations and it is probably difficult to reproduce accurately on hard materials. The difficulty of obtaining notch uniformity is, of course, more or less inherent in all impact tests, but is probably more pronounced in this case.

The Izod test requires a test-specimen 75 mm. (2.9528 in.) long with a cross-section of 10 x 10 mm. (0.3937 in.). It has a transverse 45-deg. angle, V-shaped notch 2 mm. (0.0787 in.) deep; the radius at the root of the notch being 0.25 mm. (0.0098 in.). The piece is held vertically in a vise with the lower edge of the notch at the same level as the vise edge and is broken by a swinging tup striking the test-piece on the notched side but above it. As before, the residual energy deducted from the striking energy gives the energy required for fracturing the test-piece. This energy is usually indicated on a scale by a pointer actuated by the swinging pendulum.

The Stanton test requires a test-specimen about 6.00 in. long having a 0.50-in. diameter. It has a groove cut at the center, 0.05 in. deep and 0.05 in. wide. This test-piece is held horizontally in supports 4.50 in. apart and is rotated 180 deg. between successive blows which are struck over the groove by a falling tup. The weight of this hammer, or tup, is 4.70 lb. and the height of the fall is adjustable. The number of blows required to fracture the test-specimen is recorded.

The Brinell test has often been put forward as a test in itself. It was never intended for such a use. It has a value only as related to some fundamental test such as the tensile test. It seems that this cannot be emphasized too strongly. There is some definite relation between the tensile-strength and the Brinell hardness for a given material. Similarly, attempts have been made in the past to correlate impact values and fatigue limits, yet this seems erroneous since the conditions of fracture have nothing in common. The fatigue fracture is determined by a series of slips in the material itself, while the impact fracture is propagated by a notch definitely located in the specimen. The fact, therefore, remains that a material giving a low impact-figure may not show a low fatigue-limit, but it is still relevant to state that the former material is in a notch-brittle condition.

Therefore, although it is difficult and dangerous to make definite statements in favor of any particular test the following conclusions seem to be warranted:

(1) The impact test of the Izod type is one of the most sensitive mechanical tests so far introduced to determine the design of heat-treatment given to a material

(2) This impact test shows characteristics in certain materials that the tensile tests do not indicate

(3) The impact test corresponds, along with other physical properties such as reduction of area, to the variation in heat-treatment and is therefore valuable for confirmatory purposes

(4) The impact test offers valuable evidence through observation of the fracture, relative to the thermal treatment that the material has received

Any test that brings out something new or confirms some value already known is valuable. Provided this impact-figure is not purchased at too high a price with respect to the ultimate-strength or elastic-limit, such tests have a very great value indeed.

With regard to the repeated impact, such as the Stanton, the case is somewhat different. This test is in the nature of a fatigue test and may be closely related to the fatigue limits as determined by standard machines. This is a contention, however, which has yet to be proved.—From a paper presented by J. M. Lessells before the Pittsburgh Section of the American Society for Steel Treating.

Petroleum and Automotive Engineers Discuss Common Problems

THE subject of the Thursday session of the third annual meeting of the American Petroleum Institute held at St. Louis, Dec. 5 to 9, was, *What Is Good Gasoline?* This session was arranged to afford a common meeting-ground for discussion of the fuel problem by petroleum producers and automotive engineers. In this respect it was like similar sessions held at the City of Washington 2 years ago and at Chicago a year ago. But this is almost the extent of the resemblance. Two years ago, there was no real common meeting-ground for the two groups of engineers. Though both were willing to work together on problems of mutual interest, they had no common language. A year ago, there was a much better understanding than theretofore and the foundation was laid for the joint research that formed the main topic of this year's session.

The outstanding feature of the day's meeting was the manner in which engineers from both industries discussed the fuel problem, each with a real understanding of the problems of the other. The joint research which has brought together prominent automotive and petroleum engineers on the steering committee, even if it had accomplished nothing else, has had its effect in destroying the barrier of misunderstanding that has so long prevented harmonious cooperative action on problems where mutual interests demanded joint action. But much more than this has been accomplished, as was brought out in the various papers reviewed below.

Gasoline Specifications

Another point of especial interest which was touched on by more than one of the petroleum engineers is that of the standardization of fuel quality. A standard for quality of gasoline is obviously difficult to realize; and the more the question is studied, the more difficult, in some respects, it appears to be. Hence, the petroleum industry has long opposed the adoption of such standards for very important reasons. Nevertheless, the advantages that would accrue to both industries through a uniformity of the product, particularly if the quality were properly regulated with reference to the average temperature, are so great that at least some of the prominent petroleum engineers are beginning to believe with the automotive engineers that some degree of beneficial standardization can be accomplished.

Knock Suppressors

The first paper on the program, entitled Demonstration of the Relationship of the Knock to the Gasoline Problem, and the Remedy for the Knock, was by Thomas Midgley, Jr., of the General Motors Research Corporation. Mr. Midgley described the action of the cylinder of a four-cycle engine by a demonstration model, illustrating the normal burning of the fuel and the abnormal combustion that is accompanied by the knock. He then explained the nature of the combustion phenomena as they are understood at present, and discussed the effect of the various substances which tend to suppress the knock. This was followed by a most interesting and convincing demonstration of the action of some of these knock suppressers, first in a glass tube in which was fired a mixture of acetylene and air, alternately with and without the suppressing substance, in this case diethyl telluride; and then in a Delcolite engine that was operated alternately with two samples of the same fuel, to one of which had been added a small amount of tetraethyl lead. The results in both cases were strikingly evident. In the engine demonstration he used a knock-indicator that permits the accurate quantitative measurement of the amount of knock and serves as a basis for comparison of different fuels and anti-knock dopes. This paper has been published under the title, The Chemical Control of Gaseous Detonation with Particular Reference to the Internal-Combustion Engine.[1]

The main topic of the session was introduced when Dr. S. W. Stratton, director of the Bureau of Standards, talked on The Answer So Far Given by the Joint Research Program of the Bureau of Standards and the Oil and the Automotive Industries. Dr. Stratton prefaced his report of the Bureau's research work by some pertinent remarks on the relation of research to the two industries. Not long ago the research man was looked upon as a dreamer. The present attitude of leaders of the petroleum industry is in marked contrast to this, partly as a result of recent accomplishments along research lines. It is significant that two great industries have combined in the solving of a joint research problem.

The Public and the Government

Scientists are just beginning to learn the value of combination of research facilities. For instance, the ability of the Bureau of Standards to take up the present fuel-program is due to the fact that the training of a staff of men in heat and temperature measurement was begun 20 years ago. Ten years ago this staff undertook the solution of some of the most difficult problems of the refrigerating industry, namely, the physical properties of refrigerants; at the beginning of the war part of this staff was set to work on the problems of the aircraft powerplant; and the same staff is now engaged in the present fuel-research. Without the building up of this staff through years of development, the present progress would have been impossible.

Another point that Dr. Stratton stressed was the relation between the industries and the various Government bureaus devoted to the public good. Formerly, there was much criticism of the activities of some of these Government institutions. It should be remembered that they are acting under laws passed by our own representatives. If the laws are not good, we must look to the representatives and not to the Government bureaus. But this sort of criticism is mostly a thing of the past. Industries no longer take the attitude that it is not well for the Government to know too many things. We must remember that Government bureaus are there for the use of the industry. They can be best used by looking upon them as the laboratories of the industry. The adoption of this point of view represents one of the most important changes that has taken place between the public and the Government.

Dr. Stratton then explained the conditions leading to the undertaking of this joint research by the Bureau of

[1] See the *Journal of Industrial and Engineering Chemistry*, October, 1922, p. 894.

Standards, and described the methods employed, giving a brief summary of the results obtained so far. As this material will be presented in full at the meeting of the Society to be held this month and published in a later issue of THE JOURNAL, it will not be reviewed here. The outstanding facts, however, are that four selected grades of fuel differing in volatility were very carefully studied under summer conditions in four models of passenger car, and the results showed no significant difference in fuel mileage. Special runs to determine crankcase-oil dilution showed much more dilution with the heavier fuels. A supplementary research covering low-temperature operation is in progress, the engine under test being enclosed in a refrigerating chamber where the temperature of the air and the jacket water can be kept as low as desired. Dr. Stratton then showed a few views of the buildings of the Bureau and described briefly some of the lines of industrial research undertaken by the various divisions.

R. E. Carlson, of the Bureau of Standards, who has had charge of the experimental work, explained more in detail the experimental procedure followed in the summer program, and described the equipment which has been completed for the preliminary study of the behavior of the fuels at low temperatures.

LOW-TEMPERATURE OPERATION

An engine has been mounted in one of the altitude test chambers of the Bureau of Standards, where the air and jacket temperature can be reduced to about 20 deg. below zero, and kept under control. The engine is coupled to a dynamometer outside the chamber, to control the load and speed. By using this method it is possible to make all the measurements made in the road tests and some others. Preliminary results have already been secured with this equipment and the work should be well toward completion by the time of the meeting to be held this month, when results to that date will be reported in full.

The next speaker was B. B. Bachman, of the Autocar Co., president of the Society of Automotive Engineers. He expressed the appreciation of the Society for the inauguration of this research and for the results obtained to date. For several years engineers have conducted individually experiments along these lines, but the results have been of little value because nothing has been done to assemble or coordinate them. Now the services of the most competent laboratory have been secured, and the results of many researches of incalculable value to the industry have already been secured.

Two phases of the problem are efficiency, or miles per gallon, and crankcase-oil dilution. No difference in consumption was shown by the summer test, but it is not safe to conclude that this is true in cold weather. The crankcase-dilution figures are not very conclusive, but show increases for the heavier fuels. Possibly dilution of oil on the cylinder-walls is more important than the crankcase dilution which results from it. Different models may show very different results in this respect.

Logical conclusions to be drawn are

(1) Many types of car, even cars several years old, could operate on fuel of grade *C* in summer weather. It is doubtful if grade-*D* fuel could be used with satisfactory results

(2) Seasonal variation of gasoline volatility and distribution with reference to climate would be a means of securing better utilization of the Country's fuel resources

J. T. B. Bowles, of the Tide Water Oil Co., said that the automotive engineers have complained for several years about the quality of gasoline and concluded that the refiners were not carrying on a research for its improvement. He maintained that the automotive industry had done nothing systematic to improve its own product. The joint program under discussion, which is in the hands of a neutral party, is the correct answer to these criticisms. Neither industry alone could have put over the results of tests of its own. The petroleum industry is bending all its energies to the production of acceptable fuels in sufficient quantity, but economic laws, rather than the industry, have control of the situation.

The results, so far, indicate that much higher end-point fuels can be used, even fuel *D* showing equal performance in mileage in some cases. Crankcase-dilution is the outstanding difficulty. A study of fuels having a greater percentage off at 212 deg. fahr. should be made, as it may remedy the starting and crankcase-dilution difficulties. The two industries and the general public have already benefited greatly by the results of the work of the Bureau of Standards.

Roger Chew, of the Standard Oil Co. of New Jersey, said that probably there have been as many theories as to what constitutes good gasoline as there are chemists in the petroleum industry. But extensive tests of long duration in the most diversified equipment are necessary before an answer can be attempted.

The Government specification known as Navy gasoline was the result of cooperation between the petroleum industry and the Government Interdepartmental Committee on Petroleum Specifications in meeting the demands during the war. This gasoline has proved very satisfactory, and it would be a benefit to the automotive industry if engineers could design for such a standard fuel. Unfortunately, refiners have marketed fuels both more and less volatile.

After stating the opinion that the minor questions as to chemical purity are being taken care of by the interested parties, and that the cooperative tests by the Bureau of Standards will go far in determining what the volatility should be, Mr. Chew closed his remarks with the following expression of opinion:

> A standard gasoline specification has distinct advantage in that it is in the direction of conservation, encourages the stabilization of the gasoline supply, tends toward efficiency of the engine builder, and last, but by no means least, would give as near as is humanly possible, complete satisfaction to the consumer.

ENGINE FACTORS

H. M. Crane, chairman of the Society's Research Committee, said in part:

> In the automotive industry, we have been fighting two factors in our engines, in connection with fuel. One is, to get the fuel vaporized before we attempt to ignite it. The other is to get the maximum economy that can be obtained only by using high compression-pressures. Mr. Midgley's organization was one of the first in the automotive business to visualize not only the difficulties of the engine builder but equally the problem of the oil producer.

Automotive engineers have come to realize that crude oil must be used to the best industrial advantage. Moreover, what is wasted today cannot be brought back. Mr. Crane urged particularly that the refiners make a careful study of the results of the Bureau tests, for which he himself had the highest admiration, rather than accept the interpretation placed upon them by any one man. He emphasized the importance of the carbureter-setting used by the average driver, which might be decidedly different from that used in the tests and therefore show up differ-

ences in the fuel. In concluding, he expressed an appreciation of the sincere desire on the part of the refiners to understand the difficulties of the problem facing the automotive industry and to cooperate in the solution of joint problems. He also urged greater uniformity in fuel on the basis of the results obtained in tests of Post-Office trucks run by the Bureau of Standards last summer, in which it was shown that any departure from the customary usage decreased the mileage.

T. G. Delbridge, of the Atlantic Refining Co. endorsed the sentiments of Mr. Chew. He called attention to the well-known unreliability of the judgment of the average user as to the quality of fuel. An important factor in the joint program was the preparation of the test fuels, and he explained the care given to this by the Atlantic Refining Co. The question as to what is good gasoline is not answered, but a vigorous prosecution of work along present lines will add much to our knowledge before the next annual meeting.

The afternoon session opened with a paper by O. C. Berry, of the Wheeler-Schebler Carbureter Co., who suggested finding the operating temperatures of the average automobile under different conditions in service, as a further guide in the work of the Bureau of Standards. Such measurement would show what part of the engine is at the controlling temperature as regards fuel vaporization, enable refiners to estimate the relative merits of fuels, and spur on the automotive engineer to greater efforts in the development of manifolds.

H. L. Horning, of the Waukesha Motor Co., said that the present fuel is all right as it is put out by the responsible refiners. The trouble is with the filling station and the garage men who mix in kerosene and other things and sell it for "gasoline." They are the men whom the refiners should watch. Mr. Horning stressed the importance of the Bureau of Standards tests. The equal mileages for all fuels was, he said, only a part of the story. Crankcase-oil dilution increasing with the end-point is shown even more conclusively by other tests, and it seems that dilution is the important factor.

The speaker then presented figures, obtained from the American Petroleum Institute, which showed that 69 per cent of the population and 83 per cent of the automotive equipment of the Country are located north of St. Louis. Above this latitude, temperatures below 50 deg. fahr. occur during about half the year, and below this temperature trouble occurs due to crankcase dilution. Mr. Horning then showed a curve giving the percentage of employes of the Waukesha company using their cars at different weather temperatures. Below 0 deg. relatively few cars were used.

The next speaker was R. Haskell, of the Texas Co., who called attention to the inappropriateness of the term end-point. Three main factors in successful engine operation are the fuel, the engine and the operator, and of the three the last is the most important.

Present Status of Problem

Dr. H. C. Dickinson, research manager of the Society, gave a brief review of the reasons for undertaking the joint research, and of the present status of the problem. Of the four factors that determine fuel utilization value, namely, mileage, starting ability, crankcase-dilution and carbonization, the tests have covered the first, for summer conditions; have not touched on the second or fourth; and have given some information on the third. There is much more to be done.

Dr. Frank A. Howard, of the Standard Oil Co. of New Jersey, pointed out briefly how many of the questions of 2 or 3 years ago, such as knock and mileage, have been answered at least partially, and raised the question whether crankcase-dilution is not of much less importance than it is generally supposed to be.

E. B. Phillips, of the Sinclair Refining Co., said that the refiners are "on their toes," trying to give the consumer the best gasoline they can make out of the material at hand. R. E. Wilson, of the Standard Oil Co. of Indiana, brought out the point that if carbureter-settings were made for maximum economy instead of for maximum power, differences between the test fuels would be expected, but as the average driver sets for maximum power, the reported test-results are typical.

He then discussed at some length the relation of the conventional distillation-curve to the true volatility of the gasoline and stated that the effective upper distillation-temperature is nearly proportional to the temperature of the 85-per cent point. He made a strong plea for a more rational type of volatility specification, giving only the 20 and 85-per cent points instead of a number of points as at present. This material was published in THE JOURNAL.[2]

Natural Gasoline

W. M. Welch, of the Association of Natural Gasoline Manufacturers, presented an interesting paper on the relation of natural gasoline to the fuel problem. Natural gasoline was an "outlaw" product 2 or 3 years ago. Its value is now recognized and methods of blending have been perfected so that it has come to be a most important factor in maintaining a satisfactory volatility. The amount now used is probably 10 per cent of the total gasoline marketed. The speaker suggested that the joint research be extended to include the starting characteristics of fuels blended with natural gasoline.

The Friday session of the meeting was devoted to a discussion of the heavy-fuel engine, with special emphasis on its application to motor transportation.

New Engine Types

First on the program was a paper by H. L. Doherty, which was presented by T. R. Coates, of the Toledo Edison Co. The author maintained that the petroleum industry demands the development of two types of engine in addition to those in use at present. These are

(1) A small heavy-duty engine of simple and inexpensive construction, that will serve for truck, tractor and for small stationary use. Maximum fuel-economy is not essential

(2) An engine of much larger size and of much lower initial-cost than present Diesel practice demands, for boat propulsion and for use as a stationary engine

Some of the fundamental difficulties with the combustion turbine were pointed out, and it was recommended that all proposals along this line receive careful analysis by high-class physicists before they are given financial support.

The second paper was by Harte Cooke of McIntosh & Seymour Co., who recounted some of the difficulties encountered in the course of many years' experience with injection engines designed chiefly for stationary service; difficulties mainly of a mechanical nature, such as clogging of the injection valves. The lighter oils are not satisfactory because of vaporization in the injection system, where heat is applied. Regulation of the temperature of the fuel is important to control the viscosity at the injection valve.

A point of some interest to the automotive engineer was

[2] See THE JOURNAL, January, 1922, p. 6.

brought out in a statement regarding sulphur in the fuel. It was said that sulphur up to 3½ per cent has no deleterious effect on the engine, provided a change is made to a sulphur-free fuel just before stopping; otherwise serious rusting of the cylinder walls and manifold will result after the engine has stood even for as short a time as a few hours.

F. W. Gay, of the De La Vergne Co., in the next paper also referred to sulphur in fuels as giving trouble, but stated that there is very little trouble with lubrication; less than with steam engines. Fuels with 6 or 8 per cent of sulphur had been used.

W. G. Gernandt discussed the development of a small fuel-injection engine for which he claims a speed range of from 250 to 2250 r.p.m. for a single-cylinder model, and of from 200 to 2500 r.p.m. for a four-cylinder engine of 2⅜-in. bore and 5-in. stroke.

Prof. C. E. Lucke, of the Worthington Pump & Machinery Corporation, compared the situations as regards the two distinct engine-types that have to be considered in connection with fuel utilization. The one group, represented by the automobile engine, requires fuel that is specially prepared for its use and cannot go far beyond the present range in fuel quality. This type has the weight of a great industry back of it, and can demand what it requires, but cannot readily change. The other group is represented by the heavy-oil engine in its various forms. It can use almost any kind of fuel, but has not powerful backing to facilitate its ready development. Professor Lucke went on to discuss the basic differences between the two types and pointed out that the injection type of engine is not limited as to fuel quality or as to compression pressure, both of which make high economy possible. The reasons for slower development of this type are greater engineering difficulties in design and a patent situation that has limited progress almost to the present time.

Professor Lucke discussed at some length the real progress that has been made, and concluded that the most serious problems, those of injection at high speed, proper spraying and cylinder form, have been solved. Reduction of weight and of cost are to be accomplished still, but there are precedents which show that both are possible.

Finally, the author proposed a program of concerted research to solve some of the outstanding problems of the injection engine, such as the physics of the various fuels and their behavior in small passages at high pressures and the physics of air at high pressures and temperatures, and called upon the oil industry to further these lines of development and promote as far as possible the use of the heavy-oil engine.

Max Rotter, of the Busch-Sulzer Brothers Co., presented his paper by title, but discussed Professor Lucke's paper briefly. He said that the producers of lubricants have cooperated in Diesel development because the sale of lubricants was thereby increased, but that the fuel-oil producers have not been interested because the demand for fuel oil is decreased by Diesel development. The paper itself discussed the field of application of the Diesel engine as compared with steam, and showed that its most advantageous use is for moderate-size stationary and marine plants. In very large sizes the advantages over the steam turbine are not so marked. Diesel engines can now use almost any liquid fuel, but the fuel producers have done little to standardize or to improve the situation. The makers of lubricants, on the other hand, have improved their product considerably.

A paper by Arthur West, of the Bethlehem Steel Corporation, also read by title, was followed by several brief discussions. The trend of the fuel prices and the inevitable demand for higher thermal efficiencies were topics. The author stated that the efficiencies of steam units are more limited than those of the injection-type heavy-oil engine. He was of the opinion that the costs of Diesel-engine installation are likely to decrease due to the increased quantity production and the increase in the size of the units; also that the obsolescence period of the Diesel engine should be rated as longer than that of a turbine plant.

WORLD OIL-PROBLEMS

THE aggregate acreage of all the actually producing areas in the United States is insignificant compared with the total area of the sedimentary rocks in this country. There are numerous examples in the past history of the oil industry of tests having been drilled in a region without success, the region abandoned and then, after a time, redrilled and found to contain important oil deposits. It is only reasonable to suppose that there will be additional examples of this in the future. It is a fact that the greater part of the United States has not yet been conclusively tested for oil. The mathematical chance of large untapped oil reserves in the United States is of a different character, in a positive direction, than the careful estimates already made would lead one to believe.

For a great nation to say, "Since oil is a prime necessity to us in case of war, the oil resources of our State must be developed by our own nationals," overlooks the economic and practical points that very large sums are required to develop new oil regions, and that much of the money spent in prospecting is not productive, that in case of war only developed oil fields have practical value and that, should there be a war, any developed oil field would be immediately mobilized, irrespective of whether the development was due to domestic or foreign capital. It is better economics to have the losses of development shared by foreign capital than to have the whole loss fall at home. The most complete self-interest and self-advancement says "Throw the doors wide-open."

The oil resources of the world as a whole are large beyond any calculation possible at the present time. It is now demonstrated that commercial petroleum may occur in sedimentary rocks of any age from the oldest to the youngest and in any structural position. The sedimentary rocks occupy the greater part of the earth's surface. Those found in the United States are but a small part. It is only in the United States that these sedimentary rocks have been extensively developed for oil. As the ultimate production of the United States is to the area of the sedimentary rocks of the United States, so the ultimate total oil production of the world will be to the whole area of sedimentary rocks of the world.

But, vast as the oil resources of the world are, they must be developed, if the constantly increasing needs of civilization are to be met. The United States has borne the great portion of the burden of the world's oil requirements, giving freely from its natural resources for the benefit of commerce and civilization. Unless foreign oil-fields are developed more thoroughly, the time must inevitably come when all nations will be seriously handicapped.

There are many regions in Europe, which contain promising oil territory, whose development is prevented by antiquated laws, international jealousies and local national feeling, all to the detriment of the countries concerned. The more rapidly the lessons of past mistakes are realized and corrected, the more rapidly real progress will be made. One point grows increasingly clear, far from facing world oil-exhaustion, the oil industry is at the present time merely on the threshold of a glorious future.—A. C. Veatch at American Petroleum Institute meeting.

FEDERAL ROAD MONEY

THE post-office appropriation authorizes expenditures of $190,000,000 in Federal aid for road construction. Fifty million dollars is authorized for the fiscal year beginning July 1 and $65,000,000 and $75,000,000 respectively for each of the two succeeding fiscal years. In addition, $13,000,000 is authorized for forest roads for the two fiscal years 1923 and 1924. The funds will be administered by the Secretary of Agriculture through the Bureau of Public Roads. These funds must be matched by the States and will be administered subject to the general provisions already in force.

The new legislation reduces the maximum participation on the part of the Federal Government from $20,000 to $16,250 per mile for roads constructed with the appropriation during the next fiscal year and to $15,000 a mile thereafter. Bridges over 20 ft. in span may be considered as separate projects to which this limitation does not apply. The new appropriation comes at a time when several States are nearing the limit of funds available. The authorization of funds for 3 years in advance will be of great benefit to all States in that it will permit them to lay their plans for some time ahead.

Heretofore $350,000,000 has been appropriated. On May 31 17,000 miles of road had been completed and in addition nearly 14,500 miles were under construction. Federal-aid roads in all stages total nearly 38,700 miles, costing over $287,500,000 in Federal aid. The States have appropriated approximately $380,000,000, making a total of $667,500,000. All types of road have been constructed with Federal aid to meet the varying conditions in the United States. The average cost per mile has been $17,120; Federal aid has amounted to 43 per cent of the total cost.

It is estimated that the $190,000,000 lately authorized will result in the construction of more than 25,000 miles of improved roads, which, added to the 46,000 miles that is expected to result from previous appropriations, makes a total of 71,000 miles, or nearly 40 per cent of the estimated 180,000 miles of road in the system of Federal-aid roads now being outlined.

FATIGUE OF METALS

IN a lecture delivered to the Royal Institution in 1914, the late Lord Rayleigh quoted some exasperated experimentalist to the effect that the classical theory of hydrodynamics as developed by Euler and Lagrange had no connection with the actual behavior of real fluids. The practical engineer could find, in the records of experience, grounds for an equally unrestrained indictment of the mathematical theory of elasticity. In each case the criticism, though not wholly without foundation, is too sweeping in character, and both mathematical theories have proved of very considerable service to the practician, in spite of the fact that the ideal materials, to which alone they apply without restriction, remain mere figments of the imagination.

It was inevitable that the successful development of stress analysis during the nineteenth century should have led those versed in it to underestimate the importance of the empirical factor involved in the fixation of working stresses. Fairbairn seems to have been the first to state definitely that the permissible working-stress on an iron structure is not a fixed quantity, but varies with the character of the loading. This conclusion was at the outset based on observation and experience. Although some explanation of Fairbairn's and Wohler's results had been supplied in anticipation by James Thomson's demonstration of the fact that the raising of the tension elastic-limit of a material by stretching it, involves a lowering of the compression limit, the reality of fatigue continued to be disputed even after Thomson's conclusions had been confirmed by the elaborate experiments of Bauschinger. Bridge engineers in particular were prone to attribute to the effects of impact the necessity of adopting lower unit-stresses in proportioning structures intended to carry relatively high loads.

Whatever views may be held as to the mechanism of failure by fatigue, all agree that the safe permissible stress is a function of the number of times it is repeated. The National Research Council estimates that a steam turbine may make 15,000,000,000 revolutions during its working life and a steam engine 1,000,000,000. This latter figure probably refers to land practice and to high-speed engines and can hardly be attained in sea service. Car axles are attributed a life of 360,000,000 revolutions; while main bridge-members are only called upon to undergo 2,000,000 repetitions of stress. It is of interest to note that according to this estimate rails are subjected in their lifetime to half a million repetitions of engine wheel-loads, but to 15,000,000 repetitions of the car wheel-loads.

When the phenomena of fatigue first began to be generally recognized the belief was common that the long-continued vibration caused the material to crystallize, and the older textbooks made frequent references to the fibrous character of surfaces that had failed under a steady stress. Now, however, it is known that, even in typically fibrous materials, the ultimate constitution is really crystalline, and the characteristic appearance of fatigue fractures is attributed to the progressive character of the failure.

As the result of investigations carried out at the University of Illinois by Prof. H. F. Moore and Prof. J. B. Kommers, on behalf of the National Research Council and others, some interesting conclusions have been reached. In all cases it was indicated that the endurance limit of a material can be obtained with experiments not extending beyond 10,000,000 repetitions of stress. The observers relied for the most part on rotating-bar tests, which have the advantage that very high rates of alternation of stress can be employed without risk of errors from inertia effects. The specimens were subjected to a bending-moment that was uniform over a length of about 3 in. It was found necessary to have this portion of the beam of much smaller diameter than the remainder, since without this precaution fracture always occurred at the collets, where the loads that produced the uniform bending-moment were applied. Moreover, in making the change of section, fillets of very large radius were found essential. Thus the main body of the bar being 0.4 in. in diameter and the reduced section 0.3 in., a radius of 0.5 in. at the junction was proved to be too small. Ultimately this reduced section was made as one continuous curve of 9.85-in. radius from end to end, which gave a practically parallel section about 0.20 in. long at the middle point.

A notable point brought out in these experiments is that the surface finish has an important bearing on the capacity to withstand alternating stresses. Rough-turned specimens show 20 per cent less endurance than polished ones. Test-pieces finished by grinding, though better than if merely machined, were still distinctly inferior to those with polished surfaces. There was, however, no material difference between specimens finished with rouge and those polished with emery cloth in the usual way.

One interesting result was that the ultimate-strength of the material was a somewhat more reliable index of its endurance than was the elastic-limit. The Brinell test was, however, still better; while the notched-bar tests were unsatisfactory in this regard. Of course, in this connection it must be borne in mind that, in practice, tests are made with different objects. For the most part the suitability of a material for a given purpose is estimated from the results of practical experience with it, and the purchaser accordingly merely requires some simple and rapid test to ensure that he is being supplied with the material specified, and that this is of uniform quality. Undoubtedly the notched-bar test provides a very drastic test of this latter quality.

But even 10,000,000 repetitions of the stress require several days to complete, and a less tedious plan would be highly advantageous. This appears to be afforded by Stromeyer's method, which is based upon the fact pointed out by Lord Kelvin in 1855 that a specimen is cooled by stretching it within its elastic-limit and heated by compressing it. On the other hand, any permanent distortion always heats the bar. Hence, if subjected to an alternating stress, so long as the internal work done is wholly elastic the rise of temperature produced by compression will be annulled by the reduction of temperature due to tension, and the mean temperature will remain unchanged. So soon, however, as permanent changes are produced within the metal there will be a rise of temperature.

It has been asserted that once a material has been strained beyond its elastic-limit it is ruined, a contention that is maintained in spite of the fact that plates for boiler shells are bent to form cold, and appear none the weaker for this treatment. In this connection some of the experiments made at the University of Illinois are of special interest. Suppose the endurance limit of a material, under alternating stress, has been determined, that is to say, the stress of which it will endure an unlimited number of repetitions. What will be the effect of submitting it at the outset to a limited number of higher stresses? The Illinois observers conclude that a 38-per cent over-stress applied 100 times does not permanently injure the material, but a 35-per cent over-stress applied 1000 times did apparently reduce the endurance by 4 per cent. This fact is obviously one of great interest, as it indicates that an occasional temporary overload of a machine part is of no practical importance. This conclusion may possibly lead to a revision of current ideas as to the importance of temperature stresses. It is, moreover, noteworthy that high-carbon steels were found to suffer more from occasional overloads than the softer structural materials.—*Engineering* (London).

Current Standardization Work

AS indicated in several articles appearing in the December issue of THE JOURNAL, the Standards Committee has been most active during the last few months. The status of over 50 of the 100 live subjects before the Standards Committee Divisions changed during November, which is exceptional in view of the fact that over 75 per cent of the subjects have been awaiting Subdivision action.

BRAKE-LINING

The Technical Committee of the Brake-Liners Association has recommended that the following list of inch sizes of brake-lining be accepted as an S.A.E. Standard.

1¼ x 3/32	1½ x 1/8	2 x ¼	3 x 1/8
1½ x 3/32	1¾ x 1/8	2½ x ¼	3½ x 1/8
1¾ x 3/32	2 x 1/8	3 x ¼	4 x 1/8
2 x 3/32	2½ x 1/8	4 x ¼	

It is thought that the adoption of such a revision would place the brake-lining manufacturers in a better position to object to making odd sizes such as 1⅞ x 3/32 in. The suggestion has been referred to the Brake-Lining Subdivision of the Parts and Fittings Division, which has been working on the standardization of brake-lining tests and specifications for some time in cooperation with the Bureau of Standards and manufacturers. Brake-lining testing-machines have been developed and are being used by several of the manufacturers, the tests that have been made having been carefully checked at Subdivision meetings. At the last meeting, however, the results indicated that further development work would have to be undertaken before the testing machines would meet the requirements of the industry.

CAST-IRON SPECIFICATIONS

At the last meeting of the Iron and Steel Division it was decided that the present S.A.E. Standard for iron and steel specifications should be extended to include the statement that the Division deems it inadvisable to formulate specifications for cast iron by chemical requirements, in view of the wide range of conditions to be met.

CLUTCH-RELEASE-TYPE THRUST BALL-BEARINGS

The present S.A.E. Recommended Practice for Clutch-Release-Type Thrust Ball-Bearings was adopted by the Society in July, 1921, and is printed on p. B39 of the S.A.E. HANDBOOK. Several objections to it have been raised by Society members. At the meeting of the Ball and Roller Bearings Division on Nov. 9 this subject was discussed informally. It is believed that the criticisms submitted are amply justified. The Standards Department was requested to circularize the users of this type of bearing to ascertain definitely the extent of the use of the recommended practice.

DIFFERENTIAL GEARS

The Subdivision on Differential Gears of the Axle and Wheels Division has been working in cooperation with a similar committee of the American Gear Manufacturers' Association with a view to formulating a standard for passenger-car and motor-truck differential gears and cases. A tentative recommendation has been adopted and is shown in Figs. 1 to 5.

It is felt that the present large variety of designs can be greatly reduced through the adoption of the proposed standards in practice, thereby decreasing manufacturing costs, conserving material, lessening dealers' and service-stations' necessary stock of spare-parts and greatly facilitating replacement of worn or broken parts in passenger cars and motor trucks.

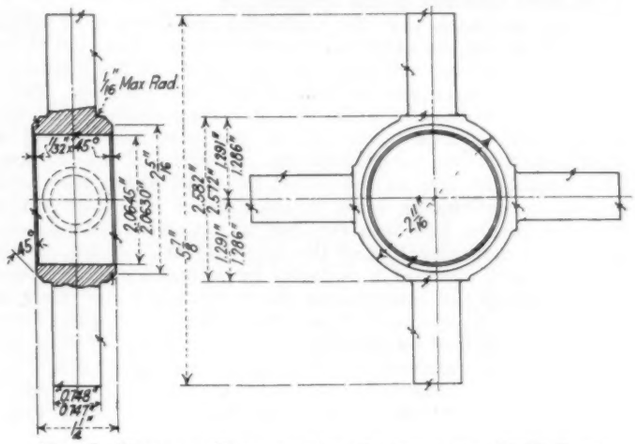


FIG. 2—PROPOSED RECOMMENDED PRACTICE FOR 11-20 RATIO GEARS

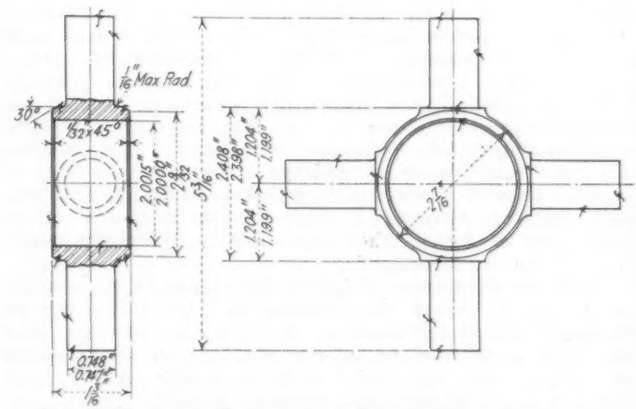
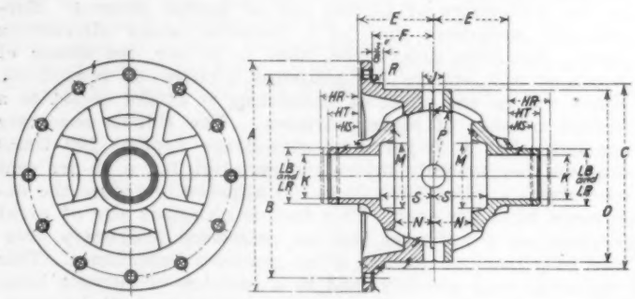


FIG. 3—PROPOSED RECOMMENDED PRACTICE FOR 10-18 RATIO GEARS



LB and *HT* for taper roller bearings
LB and *HS* for ball bearings
LR and *HR* for straight roller bearings

FIG. 1—PROPOSED RECOMMENDED PRACTICE FOR TWO-PIECE DIFFERENTIAL CASE

Differential No.	*A*	*B* ± 0.003	*C* + 0.000 − 0.002	*D*	*E*	*F* max.	*HT*	*HS*	*HR*	*J* ±0.001	*K*	*LB*	*LR*	*M*	*N* ±0.001	*P* ±0.001	*R*	*S*	Drill Holes No.	Drill Holes Diam.
11-20	8⅛	7.125	6.500	6	2⅝	2⅛	⅞	31/32	1 1/16	0.750	1 1/16	2.1669 2.1660	2.1654 2.1645	2 1/16	1.377	2.315	1/16	1⅞	12	31/32
10-18	7⅝	6.625	6.000	5½	2⅝	2⅛	⅞	31/32	1¼	0.750	1 1/16	1.9699 1.9691	1.9685 1.9677	2 1/16	1.283	2.079	1/16	1 13/16	10	31/32

Although the proposal includes many detailed dimensions, the Division has not definitely decided as to what extent detailed dimensions should be specified. Standard outside dimensions of the case would permit of interchanging or replacing cases, whereas complete standardization as tentatively proposed would tend to reduce the cost of making gears and permit individual car-owners to interchange or replace parts within the case. The recommendation proposed has been submitted to the industries generally for criticism, particularly regarding the extent to which detailed dimensions should be included. It will also be submitted as a progress report at the Standards Committee Meeting on Jan. 9 if time permits.

The standard is not intended to meet the requirements of present designs, but as a guide for future designs. It cannot be expected to meet all requirements, but should be of material value to users and makers of differentials.

Engine Numbers

At the last meeting of the Engine Division, R. S. Begg, chairman of the Passenger-Car Division, submitted a number of additional samples embodying methods for numbering engines and frames. Photographs and samples of the various methods that have been tested by the Underwriters' Laboratories, all of which were considered to have failed to meet the necessary requirements, were also submitted. It was stated that only one method of changing the numbers on each sample had been tried, but that there are probably other methods equally or possibly more effective which further study would have brought out. Upon completion of all tests by the Underwriters' Laboratories, a complete report illustrated with photographs of all samples submitted by the Society and received from other sources will be available.

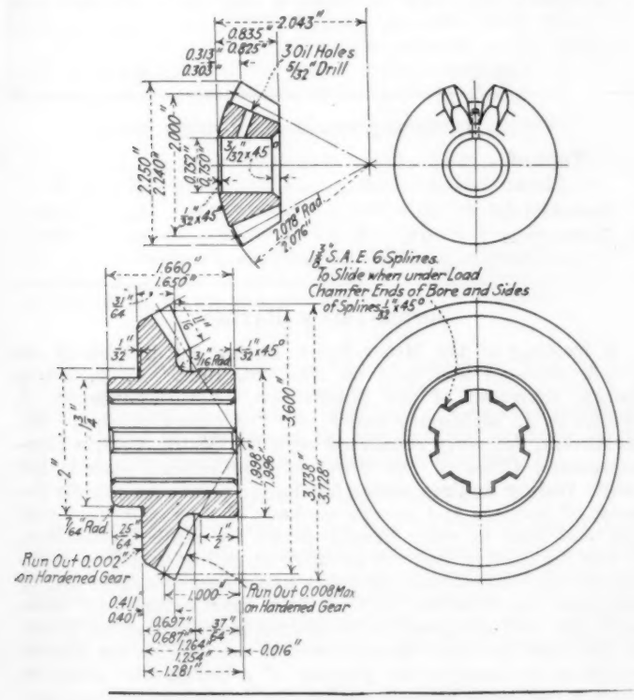


Fig. 4—Proposed Recommended Practice for 10-18 Differential Gear and Pinion

	Gear	Pinion
Number of Teeth,	18	10
Diametral Pitch	5-7	5-7
Pitch Angle, deg.-min.	60-57	29-3
Face Angle, deg.-min.	64-55	33-1
Chordal Thickness, in.	0.3093	0.3153
Corrected Pitch Depth, in.	0.1453	0.1505
Addendum, in.	0.1429	0.1429
Dedendum, in.	0.1785	0.1785
Block Angle, deg.-min.	4-57	4-57
Pressure Angle, deg.	20	20

The chordal thickness allows for 0.006-in. backlash.

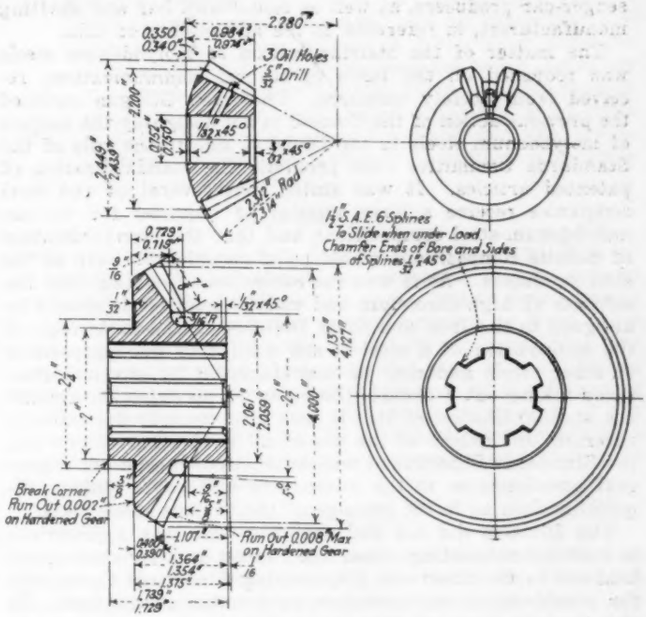


Fig. 5—Proposed Recommended Practice for 11-20 Differential Gear and Pinion

	Gear	Pinion
Number of Teeth,	20	11
Diametral Pitch	5-7	5-7
Pitch Angle, deg.-min.	61-11	28-49
Face Angle, deg.-min.	64-46	32-24
Chordal Thickness, in.	0.3080	0.3130
Corrected Pitch Depth, in.	0.1459	0.1530
Addendum, in.	0.1429	0.1429
Dedendum, in.	0.1785	0.1785
Block Angle, deg.-min.	4-22	4-22
Pressure Angle, deg.	20	20

The chordal thickness allows for 0.006-in. backlash.

Flexible Discs

The Parts and Fittings Division has undertaken the standardization of bolt-circle diameters, inside diameters, outside diameters and thicknesses of flexible discs used for propeller-shaft couplings. C. W. Spicer and E. W. Templin have been appointed a Subdivision to obtain information and prepare a report to the Division.

Flywheel Housings

An inquiry as to whether 8 or 12 holes should be specified for the S.A.E. Standard No. 5 flywheel housing flange was discussed at the last meeting of the Engine Division. As very few of the engine builders make engines requiring this size of housing, it was decided that the Standards Department should make inquiry as to what companies are using it and what mounting dimensions are considered good practice.

Iron and Steel Specifications

At the last meeting of the Iron and Steel Division it was pointed out that the present S.A.E. steel numbers are not always used by purchasers of S.A.E. standard analyses, private specification numbers being substituted. It was thought that there is absolutely no good reason for this and that the American Iron and Steel Institute should be requested to take steps to counteract the practice if possible.

The suggestion that the minimum sulphur-content for screw-stock specifications No. 1112 and 1120 be changed from 0.075 to 0.090 per cent, the other elements remaining the same, was discussed. It was decided that the the Standards Department should circularize representative steel and pas-

senger-car producers, as well as cold-drawn bar and shafting manufacturers, in reference to the advisability of this.

The matter of the standardization of molybdenum steels was reopened on the basis of several communications received from Society members. Chairman Gilligan outlined the previous action of the Council in not assigning the subject of molybdenum steels to the Division, due to the rule of the Standards Committee that prohibits the standardization of patented articles. It was stated that several of the steel companies receive a large number of inquiries for various molybdenum-steel compositions and that the standardization of definite compositions would be of considerable help to the steel producers. As it was the consensus of opinion that the subjects of high-chromium and molybdenum steels should be assigned to the Iron and Steel Division on the basis that, if the composition of a steel is not similar to the composition of other steels and may be manufactured by any manufacturer taking out a license, there can be no objection against the standardization of it. It was also thought desirable to ascertain the extent of the use of all steel compositions and the Standards Department was authorized to send out a general questionnaire to the automotive and parts makers requesting data as to the tonnage of the various steels used.

The Division did not feel that the time was appropriate to consider submitting either the carbon or alloy-steel specifications to the American Engineering Standards Committee for consideration and adoption as a national standard. It was thought, however, that the Society is the proper body to sponsor such action when it shall be considered in order.

Fuel and Lubrication Pipe-Fittings

At the last meeting of the Parts and Fittings Division the present S.A.E. Standards for Fuel and Lubrication Pipe-Fittings, pp. C46 and C47 of the S.A.E. Handbook, were discussed. The tolerance on the diameter of the hole in the extended portion for the flared-tube type that holds the tubing so as to prevent vibration, was considered with reference to possible revision. It was felt, however, that before any changes in the standard were undertaken, a careful analysis of present practice should be made, as present production of standard fittings constitutes over 50 per cent of the total production.

The Division has been working for some time also on the standardization of compression-type fittings. It is felt that any standard adopted should be based on current production.

Lubrication of Springs

Replies received to inquiries sent out recently as to the possibility of standardizing spring lubricants indicated a considerable difference of opinion as to the necessity of lubricating springs and the type of lubricant that should be used. The Springs Division has therefore decided to discontinue this subject.

Footman Loops

At the last meeting of the Passenger-Car Body Division it was suggested that a standard for footman loops would, if adopted in practice, overcome the necessity of making a large variety of designs and sizes. A type of loop which was thought satisfactory for all general applications was submitted and the Standards Department asked to circularize it for approval as a tentative recommendation of the Division. This type is shown in Fig. 6. Standard footman loops would be of value also to strap manufacturers.

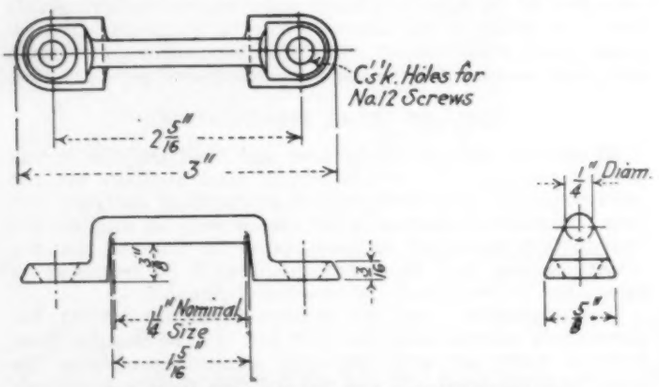


Fig. 6—Proposed Standard for Footman Loops

License-Plates and Brackets

At the last meeting of the Parts and Fittings Division, W. C. Keys submitted a report based on data received from several States indicating the largest sizes of passenger-car, motor-truck and motorcycle license-plate used. The plate dimensions proposed were based on sizes used in California, Illinois, Indiana, Michigan, New York, Ohio and Pennsylvania, in which States the majority of automobiles are registered. The report was approved in a revised form, the minimum as well as the maximum plate-lengths being specified and the holes in the plates located so that all plates for a given type of vehicle could be mounted on one size of bracket. The proposal as approved is given in Fig. 7 and is to be circulated by the Standards Department among the several States for further comment or approval before final Division action shall be taken.

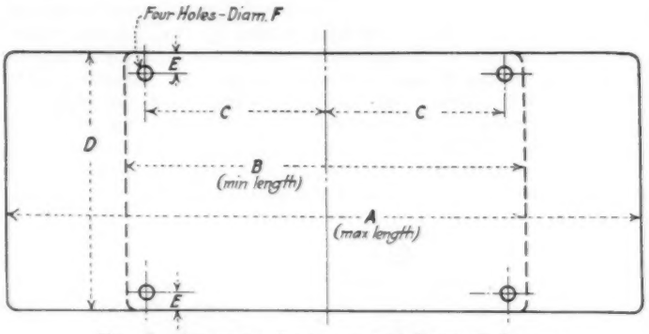


Fig. 7. Proposed Standard for License Plates

Type of Plate	A	B	C	D	E	F
Automobile[1]	16	10	4¼	6½	½	9/32
Motorcycle	8¾	5	2	4½	⅜	7/32

[1] Includes both passenger cars and motor trucks.

Motor-Truck Rating

A meeting of the Motor-Truck Rating Subdivision of the Truck Division was held on Nov. 20, 1922. A. K. Brumbaugh, chairman of the Subdivision, D. C. Fenner, A. J. Scaife, A. L. McMurtry and F. W. Fenn were present. Mr. McMurtry, formerly connected with the Motor Vehicle Commissioner's Office of the State of Connecticut, stated that public feeling against motor trucks is increasing rapidly because of careless and unsafe methods of operation, and that this will lead to more drastic action by State legislatures. It was his opinion that the problem of rating, based on regulations for brakes, steering-gears and axles is largely one of adequate maintenance in operation, particularly as these parts are well designed on recognized makes of motor truck. He felt that the most effective way to accomplish the desired results is to require the placing of a permanent plate on every motor-truck chassis giving the following information:

The gross permissible load on the front axle
The gross permissible load on the rear axle
The maximum speed
The maximum allowable stopping-distance in feet, from the maximum speed under normal conditions with the motor truck fully loaded, and applying either set of brakes independently of the other

Such plates would have to be filled out for every motor truck licensed. No trouble would be expected from new trucks made by recognized builders and with such plates attached to them they could be checked readily by inspectors on the road. This suggestion would, in Mr. McMurtry's opinion, answer practically all requirements for regulations,

leaving to the builders the determination of their motor-truck ratings, they furnishing their trucks with the plates properly stamped rather than being required to conform to an arbitrary rating. The owners of "orphan" trucks would then have to apply to the State officials for their plates and before these would be issued the truck would have to be examined by State inspectors.

After general discussion, it was decided to discontinue further consideration of the Subdivision's previous proposition for the regulation of brakes, steering-gears and axles, and to recommend tentatively that all motor trucks be equipped with plates giving the requirements necessary for safe operating conditions, and that this recommendation be circulated by the National Automobile Chamber of Commerce among its members and other truck companies in order that prompt and wide publicity might inform the public and legislators that attention is being given to providing an adequate remedy for present conditions without working a hardship on the manufacturers. The Subdivision of the Society's Standards Committee is working in cooperation with the Motor-Truck Committee of the National Automobile Chamber of Commerce. The subject of motor-truck rating is being handled by the Society as a special subject and action on it will therefore be reported direct to its Council.

Nickel-Plating

It was stated at the last meeting of the Passenger-Car Body Division that there are major problems to solve in connection with nickel-plating on steel and enameling on brass. Considerable trouble with the plating on steel peeling, especially on oversea shipments and cars or parts held in storage, has been experienced and no satisfactory way of overcoming the condition has been found. It was the consensus of opinion that nickel-plating on iron and steel, even when they are first plated with copper or zinc, should not be used except where structural conditions make it necessary to do so. The subject will be considered further.

Paints, Varnishes and Enamels

This subject was discussed at the last meeting of the Passenger-Car Body Division and considered important. It was felt that any work the Division could do would be of material assistance in bringing the makers and the users of the products to a closer understanding regarding their mutual requirements and limitations. A Subdivision is to be appointed to investigate the subject thoroughly.

Poppet Valves

The Subdivision on Poppet Valves held a meeting recently but no agreement was reached as to the extent or character of a desirable poppet-valve standard. Tentative specifications are to be considered by the various members of the Division and discussed at a subsequent meeting.

Radiator Nomenclature

Herbert Chase, writing in the Nov. 16, 1922, issue of *Automotive Industries*, has suggested definitions for various types of radiator core. These have been referred to the Radiator Division in connection with desirable revision of the present automobile nomenclature for radiator parts now printed on p. K5 of the S.A.E. Handbook. The definitions suggested by Mr. Chase follow.

Fin-and-Tube Type.—The fin-and-tube type radiator core is one made from continuous water-tubes over which the fins are fitted. The tubes have various shapes of transverse section, but are not crimped and are not of that wide, flat type which is formed by joining two metal ribbons at each edge

Ribbon Cellular Type.—The ribbon cellular type of radiator core is one made from continuous water-tubes formed by joining two metal ribbons at each edge. The walls of the tubes are usually crimped and grouped so as to form a cellular structure. In some cases a part of the cellular structure is formed by flat or crimped ribbon, which is not a part of the water-tube walls

Air-Tube Cellular Type.—The air-tube cellular type of radiator core is one made from air tubes that are nested in such a way as to form water-passages between the tubes, these passages being sealed by soldering at the ends of the tubes. In this type of core the general direction of water flow is at right angles to the tubes, but the water can flow transversely as well as vertically through the passages between the tubes

Rod-End Pins

When the present S.A.E. Standard for Rod-End Pins, p. C8 of the S.A.E. Handbook, was adopted, it was general practice to mill the heads on the pins. Due to developments in production, it is now the general practice to form these heads by upsetting. As the tolerances on upset pins are necessarily larger than those on milled pins, it is considered important to revise the present tolerances on the diameters of the holes in the rod-ends. At the last meeting of the Parts and Fittings Division, W. J. Outcalt was appointed a Subdivision to prepare a report on upset rod-end pins to fit the present rod-end holes.

Rolled-Sections

The standardization of steel and aluminum rolled-sections is one of the most important subjects before the Passenger-Car Body Division. Although a tremendous number of sections are used, it is felt that all requirements can be met by a few standard sections. Only the simpler sections, such as ovals, quarter-ovals, tees, rain-drips and gutters will be taken up at first, as the more complicated sections are more difficult to standardize. It is argued that there is need to establish standard hardness specifications for aluminum moldings because at present they vary widely in their degree of hardness.

Sheet Aluminum

At the meeting of the Passenger-Car Body Division held in November it was indicated that the present S.A.E. Standard for Sheet Aluminum, p. D121 of the S.A.E. Handbook, does not entirely answer body-makers' requirements, as it does not limit the number of sheet sizes. It was felt that a supplementary standard for sheet aluminum listing a limited number of gage thicknesses, specific sheet sizes and definite tempers, should be adopted. R. A. La Barre was appointed a Subdivision to draft a report in this connection.

Taper Fittings

The Motorboat Division expressed the view that the present standards for motorboat couplings should be revised in accordance with the standard on taper fittings for plain or slotted nuts, p. C14 of the S.A.E. Handbook. The present motorboat coupling standards specify "non-standard" nuts in many cases. It was found, however, that the present standard for taper fittings does not include fittings for the 1⅝, 1⅞ and 3¼-in. shaft sizes that are used extensively in motorboat practice. The Motorboat Division asked that the Parts and Fittings Division extend this standard accordingly. The request was referred to C. W. Spicer, of the Spicer Mfg. Corporation, who was largely responsible for the formulation of the original standard, and a report was recently submitted and approved by the members of the Division.

Metric-Type Roller Bearings

At the October, 1921, meeting of the Ball and Roller Bearings Division it was felt that there was sufficient demand for standard metric-type roller bearings that are interchangeable with wide or double-row type ball bearings to warrant the Society in continuing to publish the present roller-bearing standards on pp. C43 and C44 of the S.A.E. Handbook. As the discussion indicated that the present dimensions of the standard metric-type roller bearings are in general satisfactory it was decided to request the Roller Bearing Subdivision to continue consideration of the revisions of the present standards as to tolerances, widths and corner radii as proposed at a Subdivision meeting on Nov. 20, 1918.

TABLE 1—LIGHT-SERIES METRIC ROLLER-BEARING DIMENSIONS

Bearing No.	Inside Diameter			Outside Diameter			Width, In. Plus or Minus 0.020	Radius or Chamfer[2] C				Eccentricity Tolerances of Races, In.		Cage Clearances, In.[3]	
								Max.		Min.					
	Mm.	In.	Tolerances in Inches Plus 0.0002 Minus	Mm.	In.	Tolerances in Inches Plus 0.0000 Minus		Mm.	In.	Mm.	In.	Inner	Outer	Diam.	Width
L 04	20	0.78740	0.0004	47	1.85040	0.0006	3/16	1.5	0.06	1	0.04	0.0006	0.0012	1 11/16	0.16
L 05	25	0.98425	0.0006	52	2.04725	0.0008	3/16	1.5	0.06	1	0.04	0.0008	0.0012	1 13/16	0.16
L 06	30	1.18110	0.0006	62	2.44095	0.0008	1/4	4.0	0.16	2	0.08	0.0008	0.0012	2 3/16	0.16
L 07	35	1.37795	0.0006	72	2.83465	0.0008	7/32	4.0	0.16	2	0.08	0.0008	0.0012	2 5/16	0.16
L 08	40	1.57481	0.0006	80	3.14962	0.0008	1	4.0	0.16	2	0.08	0.0008	0.0012	2 11/16	0.16
L 09	45	1.77166	0.0006	85	3.34647	0.0008	1	4.0	0.16	2	0.08	0.0010	0.0016	3 3/16	0.20
L 10	50	1.96851	0.0006	90	3.54332	0.0008	1	4.0	0.16	2	0.08	0.0010	0.0016	3 3/16	0.20
L 11	55	2.16536	0.0006	100	3.93702	0.0008	1 1/16	5.0	0.20	3	0.12	0.0010	0.0016	3 5/8	0.20
L 12	60	2.36221	0.0006	110	4.33072	0.0008	1 3/16	5.0	0.20	3	0.12	0.0010	0.0016	3 11/16	0.20
L 13	65	2.55906	0.0006	120	4.72443	0.0008	1 3/16	5.0	0.20	3	0.12	0.0010	0.0016	4 1/16	0.20
L 14	70	2.75591	0.0006	125	4.92128	0.0008	1 1/16	5.0	0.20	3	0.12	0.0010	0.0016	4 1/16	0.24
L 15	75	2.95277	0.0006	130	5.11813	0.0008	1 1/16	5.0	0.20	3	0.12	0.0010	0.0016	4 1/16	0.24
L 16	80	3.14962	0.0006	140	5.51183	0.0008	1 5/16	5.0	0.20	3	0.12	0.0012	0.0018	5 1/16	0.24
L 17	85	3.34647	0.0007	150	5.90554	0.0012	1 3/16	5.0	0.20	3	0.12	0.0012	0.0018	5 1/16	0.24
L 18	90	3.54332	0.0007	160	6.29924	0.0012	2	5.0	0.20	3	0.12	0.0012	0.0018	5 5/8	0.28
L 19	95	3.74107	0.0007	170	6.69294	0.0012	2 1/16	5.0	0.20	3	0.12	0.0012	0.0018	6	0.28
L 20	100	3.93702	0.0007	180	7.08664	0.0012	2 3/16	5.0	0.20	3	0.12	0.0012	0.0018	6 1/16	0.32
L 21	105	4.13387	0.0007	190	7.48035	0.0012	2 1/16	5.0	0.20	3	0.12	0.0012	0.0018	6 11/16	0.32
L 22	110	4.33072	0.0007	200	7.87405	0.0012	2 3/16	5.0	0.20	3	0.12	0.0012	0.0018	7 1/16	0.35

[2]When a chamfer is used, it shall be 45 deg. from the axis of the bearing and ground true with the inside and outside diameters.
[3]The cage clearances are published as general information only.

TABLE 2—MEDIUM-SERIES METRIC ROLLER-BEARING DIMENSIONS

Bearing No.	Inside Diameter			Outside Diameter			Width, In. Plus or Minus 0.020	Radius or Chamfer[4] C				Eccentricity Tolerances of Races, In.		Cage Clearances, In.[5]	
								Max.		Min.					
	Mm.	In.	Tolerances in Inches Plus 0.0002 Minus	Mm.	In.	Tolerances in Inches Plus 0.0000 Minus		Mm.	In.	Mm.	In.	Inner	Outer	Diam.	Width
M04	20	0.78740	0.0006	52	2.04725	0.0008	3/16	1.5	0.06	1	0.04	0.0006	0.0012	1 3/4	0.20
M05	25	0.98425	0.0006	62	2.44095	0.0008	1	1.5	0.06	1	0.04	0.0008	0.0012	2 1/8	0.20
M06	30	1.18110	0.0006	72	2.83465	0.0008	1 1/16	4.0	0.16	2	0.08	0.0008	0.0012	2 1/2	0.20
M07	35	1.37795	0.0006	80	3.14962	0.0008	1 3/16	4.0	0.16	2	0.08	0.0008	0.0012	2 3/4	0.20
M08	40	1.57481	0.0006	90	3.54332	0.0008	1 7/16	4.0	0.16	2	0.08	0.0008	0.0012	3 1/8	0.20
M09	45	1.77166	0.0006	100	3.93702	0.0008	1 1/16	4.0	0.16	2	0.08	0.0010	0.0016	3 1/2	0.24
M10	50	1.96851	0.0006	110	4.33072	0.0008	1 3/16	4.0	0.16	2	0.08	0.0010	0.0016	3 1/2	0.24
M11	55	2.16536	0.0006	120	4.72443	0.0008	1 1/16	5.0	0.20	3	0.12	0.0010	0.0016	4 1/4	0.24
M12	60	2.36221	0.0006	130	5.11813	0.0008	2 1/16	5.0	0.20	3	0.12	0.0010	0.0016	4 5/8	0.24
M13	65	2.55906	0.0006	140	5.51183	0.0008	2 1/16	5.0	0.20	3	0.12	0.0010	0.0016	5	0.24
M14	70	2.75591	0.0007	150	5.90554	0.0012	2 1/16	5.0	0.20	3	0.12	0.0010	0.0016	5 1/4	0.28
M15	75	2.95277	0.0007	160	6.29924	0.0012	2 1/16	5.0	0.20	3	0.12	0.0010	0.0016	5 5/8	0.28
M16	80	3.14962	0.0007	170	6.69294	0.0012	2 1/16	5.0	0.20	3	0.12	0.0012	0.0018	6 1/8	0.28
M17	85	3.34647	0.0007	180	7.08664	0.0012	2 1/16	5.0	0.20	3	0.12	0.0012	0.0018	6 1/2	0.28
M18	90	3.54332	0.0007	190	7.48035	0.0012	2 1/16	5.0	0.20	3	0.12	0.0012	0.0018	6 1/2	0.32
M19	95	3.74017	0.0007	200	7.87405	0.0012	3 1/16	5.0	0.20	3	0.12	0.0012	0.0018	7 1/8	0.32
M20	100	3.93702	0.0007	215	8.46460	0.0012	3 1/16	5.0	0.20	3	0.12	0.0012	0.0018	7 11/16	0.32
M21	105	4.13387	0.0007	225	8.85830	0.0012	3 1/16	5.0	0.20	3	0.12	0.0012	0.0018	7 11/16	0.32
M22	110	4.33072	0.0007	240	9.44886	0.0012	3 5/16	5.0	0.20	3	0.12	0.0012	0.0018	8 1/16	0.32

[4]Where a chamfer is used, it shall be 45 deg. from the axis of the bearing and ground true with the inside and outside diameters.
[5]The cage clearances are published as general information only.

Tables 1 and 2 show the present S.A.E. Standards for light and medium series metric-type roller bearings as proposed to be revised. The tolerances for the inside and the outside diameters are as now specified in the present S.A.E. Standard. At the meeting in 1918, however, it was proposed that further consideration be given to these tolerances as they depend upon the type of bearing and its application.

Transmission Lubricating Oil

At the Joint Conference of the Interdepartment Petroleum Specifications Committee, the Advisory Committee on Petroleum Specifications and Methods of the American Petroleum Institute and the Society's Lubricants Division in the City of Washington on Nov. 13, 1922, the recommendation for transmission lubricating oils published in the November, 1922, issue of THE JOURNAL, was discussed. The principal objections were that the flash-point is too high and the pour-test value too low for the summer grade. It was also thought that the specifications might be broadened to include other grades of oil. The proposal was referred back to the Lubricants Division and was therefore discussed at a subsequent meeting of the Division. It was decided that the Standards Department should circularize the industries for further information and suggestions as to the specifications.

Sheet Steel

As the American Society for Testing Materials and the Detroit Chapter of the American Society of Steel Treaters have appointed Subcommittees on sheet-steel standardization, it has been arranged to hold joint meetings of these committees and the Iron and Steel Division's Subdivision on Sheet Steel. The dividing line between steel strip and sheet not being clearly defined, it has been decided that the Society's Sheet-Steel Subdivision shall be known as the Subdivision on Steel Sheets and Strips and cover both materials in its report.

RESEARCH REDUCES HIGHWAY COSTS

THE second annual meeting of the Advisory Board on Highway Research was held on Nov. 23 and 24 at the National Research Council headquarters in the City of Washington. This Board serves as a coordinating agency for the highway activities of the Country and is a subsidiary of the National Research Council. At the meeting the national highway-transport problem was discussed from various angles, but particular attention was given to its economic aspects.

Obviously, highways can no longer be classed as "good roads" or "bad roads." The commercialization of highway transportation of both freight and passengers in the past few years has led to a general recognition of the highway, not as ministering to the pleasure and comfort of the user, but as an essential part of a great economic whole, namely a system of transportation parallel in importance to railways and water routes.

It is a surprising fact that while the cost of highways is scarcely more than five per cent of the total cost of highway transportation, and vastly less than the cost of vehicles and their accessories, yet the recognition of highway transport as a single great economic problem has to be credited to the highway engineer, while the automotive industry, if we may judge from its public activities, has not yet even grasped the full significance of this problem.

The meeting was attended by about 100 representatives of the member societies and others, and the program occupied the full day Thursday, the 23rd. On Friday the delegates visited the experimental station of the Bureau of Public Roads at Arlington Farm, across the river from the City of Washington, and the laboratories of the Bureau of Standards. Among the most interesting exhibits at the former were a new type of impact testing-machine to determine the effect of wheel action on the road and two circular test-tracks, one of bituminous material and the other of concrete, for the study of surface wear, corrugation and the like. At the Bureau of Standards, where various automotive and highway problems are being studied, was a newly developed instrument for recording the performance of cars on the road and apparatus for the testing of tires under various conditions for power requirements and durability.

The technical session included reports from the subcommittee on Economic Theory of Highway Improvement, Structural Design of Roads, Character and Use of Road Materials, Highway-Traffic Analysis, Highway Finance and Maintenance, and from some of the member societies.

The report of the Director, Dr. W. K. Hatt, outlined the present status of highway research and described the tasks that lie ahead. It dealt particularly with the progress made by the Advisory Board on Highway Research, the needs of the Board to accomplish its purpose, the activities of the Director, and the notable researches carried out in the highway field during the year.

The report of Prof. T. R. Agg, of the Committee on the Economic Theory of Highway Improvement, was of particular interest. This covered a research project that has been in progress at Iowa State College for two or three years and is now substantially completed. The tests were designed to determine the relative fuel-consumptions and tractive-resistances of various types of vehicle on different classes of road. Most of the measurements were made by a time and distance recording device, though at least two types of accelerometer were used for part of the work.

The conclusions shown were as follows:

(1) About half the fuel consumed by a motor car or truck is required to operate the mechanism and to overcome air resistance

(2) Unevenness of surface is immediately reflected in an increase in tractive resistance

(3) A change to a better type of highway surface may easily save enough fuel to liquidate the cost of the improvement

(4) Bituminous-type roads show materially higher resistance in hot than in cold weather

(5) Rolling resistance on low-grade roads is an exceedingly variable quantity

The report of Dr. A. T. Goldbeck, of the Bureau of Public Roads, described briefly the recent experimental work at that bureau and explained the results obtained by recent tests that will permit a decrease in the cost and an increase in the strength of some types of road, notably concrete constructions. This paper gave the impression that, in contrast to the situation a year ago, design factors for highways are coming to be well understood.

The papers by H. S. Mattimore, G. E. Hamlin, J. G. McKay and W. H. Root were all of importance in their respective lines. The evening session commenced with a paper by Dr. Raymond Dodge on the Services of Psychology to Problems of Traffic Control. Professor Dodge treated the problem of traffic control in a new and interesting way, emphasizing its psychological and physiological elements to the exclusion of the mechanical ones. He discussed the mental reactions of the driver and the pedestrian under conditions of traffic and suggested a number of possible changes in the regulations to take these factors into account. In discussing the various types of highway sign, he emphasized the necessity of signs that the driver can understand instantaneously. He expressed the opinion that explanatory matter in smaller type is likely to distract attention momentarily from the main idea and thus become a source of danger.

Dr. H. C. Dickinson, of the Society of Automotive Engineers, explained briefly the organization of its Research Department and described some of the projects that are being handled by this department. He outlined some of the more interesting and urgent highway problems that concern the automotive industry, notably that of internal power-losses in vehicles and their effect on the rate of fuel-consumption; the effects of road surfaces on the maintenance cost of the vehicle; elimination of the danger of skidding; efficiency of brake action as affected by the road surface; and general questions of traffic control.

W. S. James, of the Bureau of Standards, described some of the recent work done there, including a newly perfected instrument for measuring and recording the various factors of vehicle performance on the road. The records taken include instantaneous fuel-consumption, acceleration and deceleration; instantaneous speed in steps, from which time-distance records can be deduced, as well as pressures in the intake-manifold, as an index of engine power; other pressures and temperatures at various positions, and the speed and direction of the car with reference to the air and wind. In total, 17 separate records are kept continuously and automatically in a form convenient for analysis, all but the fuel-consumption record being on a single sheet of paper.

In the absence of Major M. L. Ireland, a very brief summary of the Quartermaster Corps Tractive Resistance of Roads Research was given by Dr. Dickinson. This project has been under way for about a year and a half, and the experimental program is complete. Studies have been made of the rolling-resistance of five different vehicles with loads of from 0 to 200-per cent normal, and at speeds of from 5 to 25 m.p.h., on a variety of roads. These tests include observations with a traction dynamometer, a Wimperis recording accelerometer and a newly designed recording accelerometer of another type. Some 550 separate tests were run in all.

In addition to these runs, tests of rolling-resistance under various loads with a number of different classes of tire, pneumatic and solid, were made on a drum dynamometer in the laboratory at Sheffield Scientific School by Professor Lockwood. The results of these have already been published.

The research program also includes about 90 tests of sliding friction of tires on various classes of road surface. The results are now being compiled by a subcommittee of the Advisory Board on Highway Research.

SOCIETY MEETINGS

ANNUAL MEETING PROGRAM

TUESDAY, JAN. 9

10:30 a.m.—STANDARDS COMMITTEE SESSION

2:00 p.m.—STANDARDS COMMITTEE SESSION (Continued)

2:00 p.m.—BODY ENGINEERING SESSION

*Testing of Paints and Varnishes.....L. V. Pulsifer
Presentation Includes Actual Laboratory Demonstrations

*The Needless Hardwood Lumber Waste....F. F. Murray

8:00 p.m.—AERONAUTIC SESSION

Symposium of Five Papers on the Design of Commercial Airplanes, Followed by Prearranged Discussion

WEDNESDAY, JAN. 10.

10:00 a.m.—ANNUAL BUSINESS MEETING

Address by President Bachman
Reports of Officers and Committees
Election of Officers
A Critical Study of Modern Steering Systems,
Herbert Chase

2:00 p.m.—DETONATION SESSION

*Engine Characteristics Under High Compression,
J. H. Holloway, H. A. Huebotter and G. A. Young

The Laws Governing Detonation....Thomas Midgley, Jr.

*Testing Fuels for High-Compression Engines,
Stephen M. Lee and Stanwood W. Sparrow

2:00 p.m.—BODY ENGINEERING SESSION

Reducing the Cost of Enclosed Bodies..George J. Mercer
Testing of Leather Substitutes.....J. B. Davis

THURSDAY, JAN. 11

10:00 a.m.—RESEARCH SESSION

*Report and Results of S.A.E.-A.P.I.-N.A.C.C. Fuel-Volatility Research at Bureau of Standards
Papers by Dr. H. C. Dickinson, W. S. James and R. E. Carlson. Also Report on Automotive Highway Research

2:00 p.m.—FUEL SESSION

Further Data on the Effective Volatility of Motor-Fuels,
Robert E. Wilson and Daniel P. Barnard, 4th

*An Experimental Survey of Gasoline and Kerosene Carburetion.....C. S. Kegerreis and G. A. Young

6:30 p.m.—ANNUAL DINNER, HOTEL PENNSYLVANIA

C. F. Kettering, Toastmaster
Presentation of Collier and Mackay Aeronautical Trophies
Address by Elisha Lee, Vice-President Pennsylvania Railroad
The Motor Truck and Our Railroads—A Problem of Coordination

FRIDAY, JAN. 12

10:00 a.m.—ENGINE COOLING SESSION

*Air-Cooled Passenger-Car Engines,
S. D. Heron, C. F. Taylor and E. H. Dix, Jr.

*Cooling Capacity of Automobile Radiators,
E. H. Lockwood

*Papers marked with an asterisk are printed in full in this issue of the JOURNAL.

NEW YORK MEETING PROGRAM

Valuable Papers on Fuel Carburetion, Steering, Engine-Cooling, Airplanes and Body Design

DISCUSSION of the problems encountered in the use of fuels of comparatively low volatility will feature the technical sessions at the Annual Meeting, Jan. 9 to 12 in New York City. The results of the Fuel-Volatility Research project fostered by the Research Department of the Society will be set forth in official reports and analyses to be presented by Dr. H. C. Dickinson and members of the Bureau of Standards staff. One entire session will cover the topic of detonation and engine characteristics under high compression; another session will include a very comprehensive paper on carburetion. The detailed program of the meeting is printed on this page and it will be noted that a number of the complete papers are printed in this issue of THE JOURNAL.

THE BODY ENGINEERING SESSIONS

Two sessions on body engineering matters are a part of this year's program. The paper on Testing of Paints and Varnishes will be accompanied by laboratory demonstrations of the methods discussed in the paper. Reduction of the cost of enclosed bodies is a matter of current interest among body engineers and the paper by George J. Mercer will describe an unconventional type of construction that is sure to affect the closed body design practice of the future. Artificial leather and top material replace practically all of the metal panels in this novel construction. The testing of these materials has been a problem of some concern to body engineers; methods are unstandardized and the results cannot be uniform. This matter has received the study of many body and production men and some experience will be presented by J. B. Davis.

REFINEMENT OF STEERING MECHANISM

The paper by Herbert Chase which follows the Business Meeting makes a critical analysis of the faults in steering mechanism that result in wheel wobble, hard steering and excessive play. Designs of steering-gears are studied and tabulated data are presented for reference purposes.

The result of the ballot for 1923 officers of the Society will be announced in the Business Meeting. President Bachman will address the meeting and graphic reports of the meetings, sections and membership activities of the Society will be shown by stereopticon.

The two papers to be given in the Friday meeting are published in this issue of THE JOURNAL. Air-cooling enthusiasts will find Mr. Heron's paper a very able treatise on this type of engine and one certain to result in a valuable discussion. Professor Lockwood's paper presents quantitative data on radiator capacity in a form that makes them readily applicable in design work.

COMMERCIAL AIRPLANES TO BE CONSIDERED

It is the purpose of the Aeronautical Meeting on Tuesday evening, Jan. 9, to promote an informal discussion of the characteristics of a freight-carrying airplane. It will be a discussion of practical design features as related to the economics of operation. Four or five short papers will be read to open the discussion, but the latter is expected to consume the major part of the meeting period. Prof. E. P. Warner, V. E. Clark and W. L. Gilmore have agreed to submit introductory papers and others are expected.

All of the technical sessions will be held in the Engineer-

ing Societies Building, 29 West Thirty-ninth Street, New York City. The morning meetings will start at 10 o'clock and those in the afternoon at 2 o'clock. It is the practice of the Society to start all meetings promptly at the hour set. Members who are tardy will lose just that much of the program.

RAILROAD FARES REDUCED

Saving Will Pay Member's Dues in Many Cases

The members of the Society who will attend the New York City and Chicago meetings can purchase railroad transportation at a substantial reduction in cost. The saving will be sufficient in many cases to pay the member's annual dues to the Society. This privilege is limited to members of the Society and dependent members of their families. Please note the following directions carefully:

(1) Buy tickets to New York City between Jan. 4 and 10; buy tickets to Chicago between Jan. 25 and 31

(2) Ask the agent for a *Reduced-Fare Certificate*, not a receipt

(3) Present signed certificate for validation at S. A. E. Transportation Desk, 29 West Thirty-ninth Street in New York City, Hotel Congress lobby in Chicago, immediately upon arrival in the city

(4) Certificates will be validated Jan. 10 and 11 in New York City and Jan. 31 and Feb. 1 in Chicago. Do not plan on returning before these dates as no certificates can be validated before 250 have been received. Arrive before the dates given since no certificates can be validated at later dates

(5) Validated certificates entitle you to one-half fare returning over the same route as you came, if purchased before Jan. 17 in New York City or Feb. 7 in Chicago

(6) Reduced-fare returning tickets are not good on certain limited trains. Consult the agent regarding this

(7) It must be understood that the reduction on the return journey is *not* guaranteed, but is contingent on the attendance of not less than 250 members and dependent members of their families

(8) The Society is held accountable under the Interstate Commerce laws for any violation of the agreement limiting the reduced-fare privilege to members and dependent members of their families. Regardless of any assurance given by local railroad officials or others, absolutely no certificates will be validated unless they are presented by persons whose membership is evidenced by our records. Applicants for membership are not construed as members

The reduced-fare plan applies from stations in all States except Oregon, Washington, California, Nevada, New Mexico and Arizona. Members living in the States named may purchase tickets at regular rates to the nearest station from which reduced-fares apply and then are able to buy transportation from that station to New York City or Chicago at the reduced rate.

RAILROAD EXECUTIVE TO SPEAK

Elisha Lee Favors Coordinating Rail and Truck Transport

The Annual Dinner of the Society will be held at the Hotel Pennsylvania, Jan. 11 at 6.30 p. m. Elisha Lee, vice-president of the Pennsylvania Railroad, will address the members on The Motor Truck and Our Railroads, a Problem of Coordination. Mr. Lee is in charge of operation of the Eastern lines of the Pennsylvania System. His topic reflects his belief that the motor truck can become a valuable ally of the railroad in the transportation of the nation's goods.

C. F. Kettering has agreed to preside as toastmaster at the annual dinner, a responsibility that seems to fall upon his shoulders with annual regularity. None will regret the recurrent choice of Mr. Kettering as guide of the dinner, for this assures an ample supply of humor on the occasion of the society's most representative gathering. President Bachman and Herbert W. Alden, Presidential nominee, will address the members during the Dinner.

Application blanks for Dinner tickets reached all members in a recent issue of the *Meetings Bulletin* and also as a part of p. 551 of the December issue of THE JOURNAL. Reservations are being received in gratifying numbers and undoubtedly the attendance will equal that at the 1922 Dinner.

RADIO BROADCAST OF SPEECHES

WJZ Will Transmit Annual Dinner Program from New York City

The speeches at the Annual Dinner of the Society in New York City will be broadcasted by radio-phone from station WJZ of the Westinghouse Electric & Mfg. Co. at Newark, N. J. Microphones installed in the ball-room of the Hotel Pennsylvania will carry the words of the dinner speakers to listeners throughout a large section of the country, including most of the automotive manufacturing district. Station WJZ broadcasts on a wave-length of 360 m. and has recently been equipped to carry its programs well beyond the Mississippi River. By this popularized achievement of scientific and engineering genius it will be possible for members of the Society to enjoy the Annual Dinner program in their homes at far-distant points. Truly this is a century of marvelous engineering progress!

The exact time-schedule of the radio program will be published in an issue of the *Meetings Bulletin* that will reach the members about Jan. 3. Watch for this schedule and arrange to have your friends enjoy the Annual Dinner program with you.

NAVY AIRSHIP INSPECTED

Pennsylvania Members View Framework of the ZR1

The duralumin framework of the huge Navy rigid airship ZR1 was inspected by a group of Pennsylvania Section members at Lakehurst, N. J., Dec. 19. The spacious airship hangar, or dock as the Navy men term it, and the tall steel mooring mast were also shown to the visitors. Following the inspection, Com. R. D. Weyerbacher gave a short talk on the engineering features of the ship and explained why the Navy decided to design and construct the ZR1 in the face of the adverse public opinion resulting from the ZR2 and the Roma disasters.

The ZR1 follows German Zeppelin practice in its general design although many of the details are altered in accordance with the lessons learned from the ZR2 failure and the results of recent research work of Navy engineers. The ship will displace 2,290,000 cu. ft., will be 680 ft. long and 78 ft. wide, and will attain a maximum speed of 65 m.p.h. She will be driven by six six-cylinder 300-hp. Packard engines designed especially for airship service. These engines have attained a fuel economy of 0.43 lb. per b.hp-hr., which compares with 0.55 lb. per b.hp-hr. for the Liberty engine. Two engines drive their propellers direct, but the other four drive through a reduction gear; one pair of engines is geared so that the propeller rotation may be reversed.

Extreme care is being taken in the fabrication of every detail piece of the ship and in the final assembly, for the Navy realizes that the success or failure of the ZR1 will determine the future of lighter-than-air ships in the minds of the public. Every precaution has been taken to avoid structural failure and a board of the National Advisory Committee for Aeronautics recently approved the design as safe.

FIRE RISK MINIMIZED

Commander Weyerbacher stated that fire risk in the ZR1 has been reduced to a minimum. Some airship fires in the past have been caused by leaking oxygen being ignited by discharges of static electricity at the point of leakage. The Germans have considered the rubber present in the gas bags or the external envelope responsible for the presence of this

static electricity. They eliminated all rubber from the gas bag covering by perfecting a special cement, the formula of which has remained a secret. Since the use of this cement, German airship fires have disappeared. The Navy is working toward the perfection of a similar cementing material and eventually hopes to eliminate all rubber in the gas bags.

Helium Used for Trial Flights

The ZR1 is expected to make her first flight about July 15. Helium will be used during the trials to insure against any danger of fire from this source. The officers and engineers at the Lakehurst station do not favor the use of helium because it is roughly 30 per cent heavier than hydrogen and lowers the useful-load capacity of the ship an equal percentage. It is their belief that all airship fires have been the secondary result of some avoidable primary cause such as structural weakness or a collision.

The Pennsylvania Section has scheduled a meeting on Lubrication for the evening of Jan. 25 at the Philadelphia Engineers' Club.

SERVICE CHICAGO FEATURE

Society Meets on January 31 to Discuss the Flat-Rate System

Two papers will be presented at the Chicago Meeting of the Society setting forth the actual experience of automotive engineers who have made successful applications of the flat-rate system of conducting automotive service-stations. Don T. Hastings, formerly assistant chief engineer of the Hupp organization, will read a very complete paper on systems of repair and maintenance which he has successfully developed in his present role of Hupp dealer. Otis C. Funderburk, who is a Lincoln-Ford dealer in Boston, will describe some unusual accomplishments in applying engineering analysis to the reduction of repair costs.

Another important phase of service work will be treated by J. F. Page, of the Packard-Chicago Service organization, who will read a paper on Service Cooperation between the Manufacturer and the Dealer. Additional papers are expected on matters of interest to engineers engaged in the service branch of the industry. These papers will be read at a morning and an afternoon technical session to be held at the Congress Hotel, starting at 10 and 2 o'clock respectively.

The annual Chicago Dinner of the Society will be held at the Congress Hotel on the evening of the same day, Jan. 31, starting at 6.30 o'clock. *Godfrey H. Atkin* will be master of ceremonies in the official capacity of toastmaster. The single address of the evening will be given by a man prominent in the automotive industry. Seats for the dinner may be ordered now. Use the coupon on p. 52 of the advertising section of this issue of THE JOURNAL.

As stated in another column, reduced-fares are available to most of the members who will attend the Chicago Meeting. Be sure to avail yourself of the lower fares as an opportunity to attend the Automobile Show, the Service Meetings and the Chicago Dinner at a minimum of expense.

MECHANICS POORLY TRAINED

Repair Operations Must Be Simplified for This Reason

In an address before the Cleveland Section on Dec. 15, Paul Donavan presented a service-man's views on troubles to which automobiles are subject in the hands of the average owner.

The important theme stressed was the fact that while the engineer or production man or service-man might all be trained to the last notch, it must be remembered that a very large percentage of our automobile mechanics in the average garage are poorly trained and comparatively ignorant as regards the work they have to do. For this reason, Mr. Donavan believed that certain repairs had to be made easy for them, and he cited specific cases where simple trouble on an automobile could be cured only by very complicated operations. For example, on some cars it is necessary to dismantle a large portion of the engine before being able to grind the valves. Mr. Donavan spoke particularly of the matter of flywheel mounting on cars and the marking of valve opening and closing, claiming that the latter were very indistinct and indeterminate. He also brought up specific cases where lubrication facilities either were not provided or were at inaccessible points.

The discussion brought out some disagreements with Mr. Donavan but, on the whole, his main thesis was agreed upon. It was pointed out that it would be advisable to consider the instruction book very carefully, inasmuch as it should be made very plain, very clear and should not be written above the heads of the average mechanics. It should also be plentifully illustrated with diagrams that actually show the part of the car as it is seen by the user.

Railroads and Highways

The Railroads and the Highways, and Economical Transportation thereon, will form the subject of a paper to be read before the January meeting of the Cleveland Section by Charles O. Guernsey. This meeting will be held on the evening of Jan. 19 in the rooms of the Cleveland Engineering Society, Hotel Winton, and will be preceded by an informal dinner at 6:30 p.m. All members in the Cleveland district are invited to attend.

REMARKABLE TRUCK ECONOMY

H. L. Horning Describes Tests Claiming 100 Ton-Miles per Gal.

Tests of a motor truck that attained a fuel economy in excess of 100 ton-miles per gal. were described in papers given by H. L. Horning and James B. Fisher before a meeting of the Mid-West Section on Dec. 20 in Milwaukee. The 2½-ton truck tested was fitted with a four-cylinder engine, of 4-in. bore and 5½-in. stroke, especially designed to attain maximum thermal efficiency and economy. The total weights of the vehicle and the load ranged between 8300 and 16,000 lb. The chassis was fitted with a van body having a frontal area of approximately 40 sq. ft. A standard transmission, double-reduction axle and solid tires were used throughout the tests.

Mr. Fisher stated that the engine used had a minimum fuel-consumption of 0.48 lb. per b.hp-hr. and that the axle reduction had been selected to insure the operation of the engine through its range of greatest economy. Two routes were used for the test; one traversed typical country roads for a distance of 24 miles; the other route ran through city streets for 4 miles and included about a dozen stops. The country runs were made at a speed of 15 m.p.h. and those in the city at 12 m.p.h.

Steam Cooling System Used

A system of engine cooling was used where the cooling water is allowed to boil in the engine jacket and the steam drawn off and condensed in the radiator. Jacket-water temperatures of from 190 to 200 deg. fahr. are maintained with this system and Mr. Fisher stated that it resulted in better thermal efficiency, an increase of 8.8 ton-miles per gal. in economy, and allowed an increase in the compression-ratio because of a more uniform cooling of the cylinder. Special aluminum pistons were employed and these, in combination with a combustion-chamber form that promoted turbulence, allowed the compression-ratio to be raised to 5.5 to 1 without detonation being present. This higher compression resulted in a further increase in economy of 17.5 ton-miles per gal. The engine valves were arranged to turn slightly with each upward movement of the valve so that they were cooled more uniformly. This added another increase in economy of 5.5 ton-miles per gal.

Economy Result of Many Refinements

After all of these refinements were made, the truck with a load sufficient to bring the total weight to 16,000 lb., ran 89.6 ton-miles per gal. over the city course and 115.3 ton-miles per gal. over the country course, according to Mr.

Fisher. This remarkable economy was due to proper coordination of the units in the truck, increased thermal and mechanical efficiencies of the engine because of its higher jacket-water temperature and the employment of special forms of piston and combustion-chamber to allow the use of higher compression.

H. L. Horning presented an analysis of the causes of detonation and explained how these theories had been applied to the design of special combustion-chamber forms, piston types and spark-plug locations to reduce the possibility of detonation in the particular engine used in the economy test.

The Mid-West Section will hold a luncheon and brief meeting on Tuesday, Jan. 16, at the Chicago Engineers' Club at 12 o'clock. The committee that will choose the regular ticket of Section officers for 1923 and 1924 will be elected and reports of the various standing committees will be presented.

STEEL MANUFACTURE SHOWN

Minneapolis Section Also Hears Talk on Patterns

The complete process of steel manufacture was depicted in an educational motion picture shown to the members of the Minneapolis Section at their meeting of Dec. 6. The pictures were taken in the mills of the Interstate Iron & Steel Co. Prof. Peter Christianson of the University of Minnesota explained each of the manufacturing steps as the pictures were displayed and this added materially to the educational value of the meeting.

Design considerations in pattern and foundry practice were the subject of a talk given by R. C. Hitchcock. Examples of good and bad designs of castings were sketched and discussed, Mr. Hitchcock citing the cases causing trouble most frequently for the pattern maker and foundryman.

The next meeting of the Minneapolis Section has been arranged for Jan. 3. H. L. Bollum, president of the Twin City Motorbus Co., will speak on the Operation of City and Country Bus Service in Minnesota. A. J. Scaife, consulting engineer of the White Motor Co., is also expected to contribute a paper.

PISTON-RING DEVELOPMENT

Forms Topic of Washington Section Meeting

Multiple-piece piston-rings are losing favor rapidly, according to Allen W. Morton, who addressed the meeting of the Washington Section on Dec. 8. This is because of their greater liability to breakage, poor wearing quality in service and difficulty of installation. Individually cast rings are superior to pot-cast rings because of their uniformity, closer grained metallic structure, longer life and greater resiliency. Foundry practice of molding, melting and pouring is of greater importance than the chemical composition of the iron. Mr. Morton discussed methods of producing tension in rings and favored the practice that effects the result by artificial means such as peening, believing this assures better control of the final pressures on the cylinder-wall. He described a method of determining cylinder-wall pressures of rings by lapping a sample ring to produce rapid wear and then measuring the amount of ring wear at numerous points on the periphery. The importance of ring fit in the piston groove was stressed as a means of preventing oil-pumping.

It was brought out in the discussion that the present tolerance of from 0.001 to 0.005 in. in the width of rings might be confined to 0.0005 in. as a maximum and 0.0002 in. as a minimum. This, of course, would only be practical provided the piston manufacturers could standardize the width of the ring-grooves and confine their tolerances to a lesser limit than they do now. It was also brought out by Mr. Morton in discussion that nearly 90 per cent of the finished rings are now lathe turned, and hardly more than 10 or 15 per cent ground, whereas just the reverse of this was true 4 or 5 years ago. Mr. Morton expressed a preference for the turned rather than the ground ring. He had found oil-pumping less frequent in engines where well seasoned gray-iron rings of the plain type were used and properly treated to secure a uniform tension, best results being secured with three rings at the top of the piston and a fourth ring in the skirt.

Motor-Truck Transportation

The next meeting of the Washington Section will be held at the Cosmos Club on Jan. 5. Major Brainerd Taylor will read a paper on the Emergency Motor-Transport Plan in the United States Army. A second paper will be read by W. P. Eno on the Engineering Aspects of Traffic.

BOSTON MEETING ON SERVICE

Packard Executives Address New England Section of Society

Flat-rate service charges were discussed very fully at the New England Section meeting in Boston on Dec. 21. Knox Brown, service manager of the Boston Packard service-station, gave the principal paper. A representative of the Packard factory participated in the program and outlined the manner in which the factory supported the dealers in their efforts to render satisfactory maintenance service to the car-owner.

Recent Development of the Induction Magneto will be the topic of the next meeting of the New England Section on Jan. 25. The paper of the evening will be read by E. A. Robertson, chief engineer of the Splitdorf Electrical Co.

SCIENTIFIC WORK IMPORTANT

Physicists to Be Strong Influence in Automotive Development

Scientists and physicists will influence a large part of the automotive development of the future. This is the belief of J. H. Hunt, electrical engineer of the General Motors Research Corporation, who addressed the Indiana Section on Dec. 14. Mr. Hunt defined pure science as contrasted with engineering. The scientist works as an individual, his assistants being of a type that does not compare in mentality or training to the man himself. The scientist studies his problem, the apparatus to be used and frequently invents mathe-

Schedule of Sections Meetings

JANUARY

3—Minneapolis Section—Operation of City and Country Bus Service—H. L. Bollum
5—Washington Section—Emergency Motor-Transport Plan in the Army—Major Brainerd Taylor
The Engineering Aspects of Traffic—W. P. Eno
16—Mid-West Section—Business Meeting and Luncheon
18—Buffalo Section—Informal Dinner
19—Cleveland Section—Railroad and Highway Transportation—Charles O. Guernsey
25—New England Section—Recent Developments of the Induction Magneto—E. A. Robertson
25—Pennsylvania Section—Lubrication Meeting
Dayton Section—No meeting in January
Detroit Section—No meeting in January
Indiana Section—No meeting in January
Metropolitan Section—No meeting in January

matics to aid in the solution. He does not count the cost in time or money; the fact that it may take his lifetime to solve a problem does not discourage him. The scientist publishes everything he does just as soon as he is certain of the accuracy of his results; he is known by what he publishes and is very certain of his statements and conclusions.

The work of the engineer, in contrast, is done as part of an organization. He seldom gets credit for his work, especially during his early years. The engineer must count the cost in time and money; he is required to secure a result in a specified time. When an engineer has perfected a mechanism he is seldom at liberty to publish its details, for such publication is controlled by the patent and sales executives.

Mr. Hunt predicted that scientists would develop new and better engineering materials in the future. He believed that the engineers will adopt the scientist's method of work in the next decade. The importance of testing details rather than assembled units will be appreciated. When one wishes to study a carbureter, it will be tested individually as a carbureter, not as a part of the entire engine system. There will be constantly increased cooperation between the engineer and the scientist, fewer accidental discoveries will occur and automotive engineering will be more generally based on scientific investigation and analysis.

The Indiana Section will not hold a meeting in January because of the possible conflict with the national meetings of the Society and the shows.

FUEL TESTS DESCRIBED

The Detroit Section was addressed by H. M. Rugg of the Dodge Bros. engineering staff at its meeting on Dec. 21. Mr. Rugg described the methods being employed by the Bureau of Standards in making the series of fuel-volatility tests for the cooperative fuel-research program fostered by the Society. Mr. Rugg was assigned by Dodge Bros. to assist in the work at the City of Washington and acted as observer on a majority of the tests.

The Detroit Section has announced that no engineering meeting will be held in January. There is a possibility that a meeting will be arranged on a subject of joint interest to production and designing engineers.

BETTER ENGINEERING NEEDED

Professor Upton of Cornell University Urges Lighter Weight Design

The Buffalo Section members enjoyed a very interesting paper by Prof. G. B. Upton on the occasion of the Section meeting of Dec. 15. Professor Upton's paper suggested means of applying the results of testing materials to the design of lighter automobile chassis. He believed that the body structure should be made an integral part of the chassis assembly in order that the roof or top would add to the rigidity of the entire unit. The high-speed engine will not be the ultimate type, in Professor Upton's opinion. He favors the use of carbon-steel in automobile frames since the primary requisite is rigidity and frames of deep section are amply strong without employing expensive alloy-steels.

The discussion of the paper brought out several opinions that are worthy of note. David Fergusson concurred with Professor Upton's belief that carbon-steel can be substituted for alloy-steels in many places without sacrificing strength to a serious degree. He cited the case of crankshafts where diameters have been steadily increased to resist torsional vibration with a consequent increase in their strength that will allow the substitution of carbon-steel for the expensive alloy-steels previously required. Otto M. Burkhardt was not certain that all tops could be made permanent, for there is still a demand for open bodies with folding tops in the warmer regions of the world.

The Buffalo Section officers are arranging an informal dinner for all members in the Buffalo district to be held during January. The date is set for Jan. 18 and other details will be mailed to the members later.

AIR-COOLED ENGINES

Franklin Engineer Presents Design and Performance Data

The important engineering features of the Franklin air-cooled engine were the subject of a talk given by C. P. Grimes, research engineer of the H. H. Franklin Mfg. Co., before the Metropolitan Section on Dec. 14. Mr. Grimes explained in detail the construction of each of the important engine and chassis units, emphasizing in each instance the points peculiar to the air-cooled design.

The cylinders of the engine have a total fin radiation surface of 650 sq. in., there being a total of 52 sheet steel fins, each 5 in. long, cast into the cylinder barrel. The length, the spacing and the number of the fins were arrived at after much experimenting. Mr. Grimes warned that there could be more and longer fins inserted but there would not be sufficient air passing by them to increase the cooling effect. Passages must be made large enough to avoid clogging with insects, dirt or other obstructions often encountered in cross-country driving.

The compression in the Franklin engine is about 72 lb. per sq. in. at the period of maximum torque; the brake mean effective pressure reaches a maximum of 77 lb. per sq. in. The air passing over the cylinders rises in temperature about 100 deg. fahr., keeping the cylinder-heads below a maximum temperature of 400 deg. fahr.

H. M. Crane remarked in the discussion that the valves in air-cooled engines are less accessible than in the water-cooled types; it is almost essential that the heads be integral with the cylinder barrel, necessitating the removal of the entire cylinder to clean carbon deposits and grind the valves.

No meeting of the Metropolitan Section will be held during January because of the national meeting of the Society in New York City, Jan. 9 to 12. The series of meetings will resume in February.

CARBURETION DAYTON SUBJECT

The meeting of the Dayton Section on Dec. 19 was addressed by P. S. Tice of the Stewart-Warner Corporation. His topic was Utilization of Low-Grade Fuels in Automotive Engineering. Mr. Tice read a well-prepared paper that aroused a lively discussion, in which members of the McCook Field and General Motors Research organizations participated. Thos. Midgley, Jr., presented theories in explanation of the detonation phenomena, basing them on the results of his latest research work.

The Dayton Section has no meeting scheduled for the month of January.

Applicants *for* Membership

The applications for membership received between Nov. 15 and Dec. 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ALLISON, FRED, electrical engineer and superintendent of power, Ford Motor Co., *Detroit.*

ANDERSON, CARL C., assistant comptroller, Republic Motor Truck Co., *Alma, Mich.*

BARTUSKA, JOHN F., engineer, Maremont Mfg. Co., *Chicago.*

BELLOWS, LYMAN H. Detroit sales office, Walden-Worcester, Inc., *Worcester, Mass.*

BUTZ, JOSEPH E., research engineer, General Motors Research Corporation, *Dayton, Ohio.*

CAMMANN, OSWALD, JR., engineer, Davis Sewing Machine Co., *Dayton, Ohio.*

CHRISTIE, M. G., wing commander, Royal Air Force, British Embassy, *City of Washington.*

COLLINS, W. D., assistant engineer, H. C. Stutz Motor Car Co., *Indianapolis.*

CORNFORTH, R. GARDNER, chief engineer, Nice Ball Bearing Co., *Philadelphia.*

CRAM, RAMON S., vice-president and general manager, Ramon S. Cram Industries Co., *Columbus, Ohio.*

DE LA VERGNE MACHINE CO., *New York City.*

DOWNES, GEORGE Q., JR., student, Ohio State University, *Columbus, Ohio.*

ELLEMAN, THOMAS H., research engineer, General Motors Research Corporation, *Dayton, Ohio.*

ERNST, OTTO, dentist and inventor, *Larchmont, N. Y.*

FAIRCHILD, JOHN GUTHRIE, assistant professor, Rensselaer Polytechnic Institute, *Troy, N. Y.*

FAY, CHARLES R., production supervisor, Weidely Motors Co., *Indianapolis.*

FISHER, R. B., student engineer, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

FORBES, MYRON EDSON, vice-president and general manager, Pierce-Arrow Motor Car Co., *Buffalo.*

FORDHAM, THOMAS B., superintendent, Delco Light Co., *Dayton, Ohio.*

FREYMAN, ALEXANDER, draftsman, Western Electric Co., *New York City.*

GEISE, JOHN HARLIN, engineer, Aeromarine Plane & Motor Co., *Keyport, N. J.*

GREENE, LOUIS, student, Polytechnic Institute of Brooklyn, *Brooklyn, N. Y.*

GUGLER, JULIUS H., vice-president, Globe Electric Co., *Milwaukee.*

GUTMAN, HARRY, design engineer, Advance-Rumely Co., *LaPorte, Ind.*

HARDIMAN, FRANK M., assistant to president, Harrison Radiator Corporation, *Lockport, N. Y.*

HAUTZENROEDER, RICHARD WAGNER, Aultman-Taylor Machinery Co., *Mansfield, Ohio.*

HINCHMAN, HERMAN H., assistant chief and road inspector, Lexington Motor Co., *Connersville, Ind.*

INGALS, E. FLETCHER, 2ND, student, Massachusetts Institute of Technology, *Cambridge, Mass.*

JONES, AUGUST F., student, Cornell University, *Ithaca, N. Y.*

KEDZIE, LAURENCE FYFE, manager of sales, eastern division, automotive department, Dahlstrom Metallic Door Co., *Jamestown, N. Y.*

LACEY, ARTHUR H., senior member, Arthur H. Lacey Co., *Los Angeles, Cal.*

LANDES, JAMES U., engineer, Standard Oil Co., *Portland, Ore.*

LATIMORE, D. S., student, Alabama Polytechnic Institute, *Auburn, Ala.*

LEWELLEN, A. R., electrical engineer, Chevrolet Motor Co., *Detroit.*

MCALLISTER, RALPH HAMMOND, detail draftsman, *Urbana, Ill.*

MANBECK, PARK D., brush sales engineer, National Carbon Co., Inc., *Cleveland.*

MASON, OSCAR E., foreman of block testing, International Motor Co., *Plainfield, N. J.*

MONTANYA, H. L., inspector and salvage supervisor, Moon Motor Car Co., *St. Louis.*

MOORE, IRA A., in charge of repair department, Reliance Mfg. Co., *Cedar Rapids, Iowa.*

MORGENSTERN, GEORGE H., general sales manager, automobile division, Bullard Machine Tool Co., *Bridgeport, Conn.*

MORRISON, J. B., assistant chief engineer, Durant Motors of Canada, *Leaside, Ont., Canada.*

MORTON, ANSEL N., department head, International Motor Co., *Plainfield, N. J.*

MURAKAMI, YOSHI-AKI, works manager, Ishikawajima Shipbuilding & Engineering Co., Ltd., Tokyo, Japan.

NAFZGER, EDWIN, designing engineer, Koppel Industrial Car & Equipment Co., *Koppel, Pa.*

PARCELL, EARL W., designing engineer, Fairbanks, Morse & Co., *Beloit, Wis.*

PARKER, GEORGE ARNOLD, designing engineer and draftsman, Witteman Aircraft Corporation, *Hasbrouck Heights, N. J.*

PECK, FRANK, superintendent, Iowa State Auto and Tractor School, *Sioux City, Iowa.*

PELTON, AYLMER E., draftsman, Union Construction Co., *Oakland, Cal.*

PLEUTHNER, CHARLES, passenger-car engineer, Pierce-Arrow Motor Car Co., *Buffalo.*

POLLITT, CAPT. ALLEN V., shop superintendent, Quartermaster Corps, *Camp Normoyle, Tex.*

RENOLD, HANS, INC., *New York City.*

SAMPLE, SAM STEELE, student, University of Illinois, *Urbana, Ill.*

SAWYER, GEORGE W., motor-vehicle supervisor, Detroit division, Sinclair Refining Co., *Chicago.*

SCHNEPEL, HERBERT, layout draftsman, Nordyke & Marmon Co., *Indianapolis.*

SHOEMAKER, FRED R., student, University of Illinois, *Urbana, Ill.*

SOKAL, EDWARD, vice-president and technical manager, Katalite Corporation, *New York City.*

SPARKS, CLIFFORD M., sales engineer, Sparks-Withington Co., *Jackson, Mich.*

SWENSON, HJALMAR S., aviation-engine mechanic, Air Mail Service, *Rock Springs, Wyo.*

TANAKA, TSUNESABURO, superintendent of body department, Yanese Automobile Co., Ltd., *Tokyo, Japan.*

TODD, CHARLES RUSSELL, assistant service manager, Olds Motor Works, *Lansing, Mich.*

TREVELYAN, HARRY A., assistant supervisor of inspection, Studebaker Corporation of America, *Detroit.*

USHIODA, SEIKICHI, student, Cornell University, *Ithaca, N. Y.*

VOLK, ALBERT, engineer, G. Elias & Bro., *Buffalo.*

WALTER, CHARLES TAYLOR, experimental engineer, Swift & Co., *Chicago.*

WHITEHEAD, J. FRAZER, secretary and treasurer, Kales Stamping Co., *Detroit.*

WILLIAMS, HOMER R., sales engineer, Aluminum Co. of America, *Detroit.*

WOLLERING, MAX F., vice-president, Studebaker Corporation of America, *Detroit.*

YACKEY, HAROLD HILGARD, student, University of Illinois, *Urbana, Ill.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Nov. 10 and Dec. 9, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ATANGAN, TOMAS T. (J) 3800 Grand Boulevard, *Chicago.*

BURNELL, E. (M) assistant to president, Overman Cushion Tire Co., 250 West 54th Street, *New York City.*

CIAFFONE, JAMES V. (J) president, Engineering Automobile School, 474 Steinway Avenue, *Long Island City, N. Y.*

DUNSMORE, R. B. (M) manager of plant, Security Mfg. Co., *Los Angeles, Cal.*, (mail) 4119 South Western Avenue.

FOISY, GEORGE A. (M) designing engineer, U. S. Cartridge Co., *Lowell, Mass.*, (mail) 1084 Bridge Street.

GILBERT, H. HOYT (M) research engineer, Cadillac Motor Car Co., Detroit, (mail) 171 Florence Avenue, *Highland Park, Mich.*

GLICK, PHILLIP P. (J) inspector, Moon Motor Car Co., *St. Louis*, (mail) 4145 Lindell Boulevard.

GOTTSCHALK, V. H. (M) research engineer, Society of Automotive Engineers, Inc., 29 West 39th Street, *New York City.*

HERMAN, F. P. (A) manager of factory sales, Houdaille Co., *Buffalo*, (mail) 1400 West Avenue.

HOWE, EARL E. (J) Baker Wheel & Rim Co., 659 Book Building, *Detroit.*

IRELAND, MAJOR MARK L. (S M) Quartermaster Corps, (mail) Room 10-219, Massachusetts Institute of Technology, *Cambridge, Mass.*

JACKSON, ELWELL RANSOM (M) automotive engineer, Rock Island Arsenal, Rock Island, Ill., (mail) 804 Kirkwood Boulevard, *Davenport, Iowa.*

KENNEDY, CAPT. GRAFTON S. (S M) Ordnance Department, Aberdeen Proving Ground, Aberdeen, Md., (mail) care Quadrangle Club, *Chicago.*

LEA, R. W. (A) executive vice-president and general manager, Stephens Motor Car Co., Inc., *Moline, Ill.*

MACKENZIE, RUSSELL E. (A) carburetor engineer, Detroit Lubricator Co., *Detroit*, (mail) 331 Chandler Avenue.

MACKIE, NORMAN S. (A) treasurer and general manager, Electrocar Corporation, 501 Fifth Avenue, *New York City.*

MACVICHIE, DONALD (A) industrial engineer, Motor Transit Co., Los Angeles, (mail) 1248 Gordon Street, *Hollywood, Cal.*

OHTSUKA, CAPT. N. (F M) Imperial Japanese Artillery, Tokyo Military Arsenal, Japan, (mail) care of Mitsui & Co., 65 Broadway, *New York City.*

ORR, ERNEST M. (M) assistant general manager, Oakland Motor Car Co., *Pontiac, Mich.*

PRICE, LEONARD C. (J) student, Cornell University, *Ithaca, N. Y.*, (mail) 102 Oxford Road.

PRODOEHL, H. G. (A) in charge of tool design department, tractor works, International Harvester Co., *Chicago*, (mail) 2135 Coulter Street.

RICK, EDWARD L. (A) engineer, H. S. Honigsberg & Co., Peking, China, (mail) 671 East 49th Street, *Los Angeles.*

SIMPSON, ALBERT F. (A) field representative in charge of Mexico and Central America, White Co., Paseo de la Reforma, No. 122, *Mexico, D. F., Mexico.*

SMITH, LEOPOLD J. (A) draftsman, American Railway Express Co., New York City, (mail) 726 Jassamine Way, *Grantwood, N. J.*

SMITH, T. D. (A) sales manager, Harvey Electric Co., Chicago, (mail) 1433 Rollins Road, *Toledo.*

STANLEY, THOMAS R. (A) general manager, Oil City Engine & Power Co., 401 West Third Street, *Oil City, Pa.*

STOREY, EDWIN F. (J) layout draftsman, Paige-Detroit Motor Car Co., *Detroit*, (mail) 3010 Harding Avenue.

SUTHERLAND, J. D. (A) general sales manager, Wyman-Gordon Co., Worcester, Mass., (mail) 941 Society of Savings Building, *Cleveland.*

TILLEY, NORMAN NEVIL (S M) aeronautical mechanical engineer, engineering division, Air Service, McCook Field, *Dayton, Ohio*, (mail) 2046 Ravenwood Avenue.

TITCHENER, WALTER E. (M) production manager and engineer, Cortland forging division, Brewer-Titchener Corporation, *Cortland, N. Y.*

UPSTILL, EDGAR D. (J) experimental assistant, White Motor Co., Cleveland, (mail) 2683 Coventry Road, *Shaker Heights, Cleveland.*

WAHLBORG, EWALD G. (M) body draftsman and designer, Zeder-Skelton-Breer Engineering Co., Newark, N. J., (mail) 136 Coe Avenue, *Elizabeth, N. J.*

WOOLSON, HARRY THURBER (M) engineer, Zeder Motor Car Co., Newark, N. J., (mail) 8 Laurel Avenue, *Summit, N. J.*